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EXPERIMENTAL ENGINEERING

PREFACE TO SEVENTH EDITION.

THIS edition of "Experimental Engineering," as compared with the earlier editions, has been rewritten and enlarged to such an extent that it may be considered substantially a new book.

The work of revision has been done principally by Professor H. Diederichs, whose name appears on the title-page, in connection with that of the original author.

"Experimental Engineering" was first written by Professor Carpenter in 1892; its object, in brief, was to serve as a textbook for students in mechanical engineering and as a reference book for engineers on the subject of testing. The book was an outgrowth of an earlier work entitled "Notes on Mechanical Laboratory Practice," published in 1891.

The work was received with general favor and it has been extensively used in colleges as a textbook and by engineers as a reference book on the subjects of which it treats. More than 10,000 copies have been published and sold in this country and abroad. It has been revised from time to time, and a considerable amount of new material has been added since the publication of the first edition. The first edition, published in 1892, contained about 550 pages, while the sixth edition, published in 1906, contained nearly 850 pages.

The rapid development in mechanical engineering and the numerous improvements in the methods of investigation and in the apparatus for conducting investigations have been such as to make it desirable to rewrite and republish the entire work. Fortunately the original author was able to secure the coöperation of his colleague, Professor Diederichs, in this work, and to induce him to assume charge of the revision. Professor Diederichs, from his training and experience, and also from his acquaintance with European practice, was especially well qualified for this task. It is believed that the new edition will be found to give practically all the information required by the mechanical engineer for testing in any field of work which has received consideration from engineering or scientific societies.

In the revision the general plan of the original work has been followed. Some portions of the old work have been cut out as obsolete or practically unnecessary, and a large amount of new matter has been added. Every attempt has been made to bring the new edition up to date in every department which it covers.

The field embraced by the work is large, since it covers nearly all departments of testing, with the exception of electrical; and although every effort was made to state all explanations and directions as concisely as possible, consistent with clearness, and to eliminate all apparently unnecessary matter, it was not found possible to keep the size of the book within the limits of the preceding edition. In spite of every effort of this kind, the work is over 200 pages larger than the sixth edition. The principal increase in size is due to the extended treatment given the subjects relating to gas engines and producers, steam turbines, refrigerating and hydraulic machinery, the first two of which are comparatively late developments. By the use, however, of a thinner grade of paper, the thickness of the book will not be greater than that of the earlier edition containing about 250 pages less, so that the reader will find that the increase in the number of pages of the book to 1100 will not make it inconvenient to handle or consult, nor will it detract from its use as a textbook.

The book is intended chiefly for use in engineering laboratories and presents information which the experience of the authors has shown to be necessary to carry out experiments intelligently and without great loss of time on the part of students. For this purpose it gives a brief statement of the theoretical principles involved in connection with each experiment, with references, where necessary, to more complete demonstrations, short descriptions of the various classes of engineering apparatus or machinery, a full statement of methods of testing, and of preparing reports. For a few cases, where references cannot be readily given, demonstrations of the fundamental principles are given in full.

This work deals principally with educational methods, the use of apparatus, and the preparation required for making a skilled observer. It is believed, however, that the volume will not be without value as a reference book to the consulting and practising engineer, since it contains in a single volume the principal standard methods which have from time to time been adopted by various engineering societies for the testing of materials, engines, and other machinery, and an extensive series of tables useful in computing results. It also contains a description of the apparatus required in testing and directions for taking data and deducing results in engineering experiments as applied in nearly every branch of the art.

An attempt has been made, by dividing the book into several chapters of moderate length and making the paragraphs short, to make references to the book easy to those who care to consult it.

The full list of subjects treated in the book is given in the Table of Contents, which immediately follows the Preface. Some of the more important divisions of the work are as follows:

Experimental Methods of Investigation.
Reduction of Experimental Data. The Slide Rule; the Planimeter.
Strength of Materials, Including General Formulæ, Testing Machines, and Methods of Testing.
Methods and Instruments for Measurement of Pressure.
Methods and Instruments for Measuring Temperature.
Methods and Instruments for Measuring Speed.
Friction and the Testing of Lubricants.
Measurement and Transmission of Power.
Heat and Properties of Gases and Vapors.
Measurement of Liquids, Gases and Vapors.
Combustion and Fuels.
Methods of Determining the Amount of Moisture in Steam.
The Engine Indicator.
The Indicator Diagram.
The Testing of Steam Boilers.
The Testing of Steam Engines, Pumping Engines and Locomotives.
Steam Turbines.
Injectors.
Gas Engines and Gas Producers.
Hot Air Engines.
Air Compressing Machinery.
Mechanical Refrigeration.
Hydraulic Machinery.

The authors desire to express their thanks for assistance in the preparation of this edition to Assistant Professor G. B. Upton, who has revised and rewritten the portion of the book relating to Materials, and afforded valuable assistance in other portions of the book. They also desire to express their thanks for the help rendered by various colleagues and assistants in Sibley College, particularly to Professor C. F. Hirshfeld, Mr. A. G. Kessler, and Mr. G. L. Current.

The subject-matter of the book comes, of course, from a great variety of sources. It has been the aim in every case to give due credit, in the body of the book, to the authorities from whom information has been obtained in connection with the subject-matter under discussion.

R. C. C.

ITHACA, N. Y., *October, 1911.*

TABLE OF CONTENTS

CHAPTER I.

INTRODUCTORY.

ARTICLES PAGE
1. Objects of Engineering Experiments 1
2. Relation of Theory to Experiment 2
3. Method of Investigation 2
4. Classification of Experiments 3
5. Efficiency Tests 3
6. Classification of Experimental Errors 4
7. Probability of Errors 5
8. Errors of Single Observations 5
9. Example 6
10. Combination of Errors 7
11. Deduction of Empirical Formulæ 7
12. Rules and Formulæ for Approximate Calculation 8
13. The Rejection of Doubtful Observations 10
14. When to Neglect Errors 10
15. Accuracy of Numerical Calculations 11
16. Methods of Representing Experiments Graphically 12
17. Autographic Diagrams 14

CHAPTER II.

APPARATUS FOR REDUCTION OF EXPERIMENTAL DATA AND FOR ACCURATE MEASUREMENT.

18. The Slide Rule 16
19. Directions for Using the Slide Rule 17
20. The Vernier 20
21. The Polar Planimeter 21
22. Theory of the Instrument 23
23. Forms of Polar Planimeters 28
24. The Mean Ordinate by the Polar Planimeter 29
25. The Suspended Planimeter 31
26. The Coffin Planimeter and Averaging Instrument 31
27. Theory of the Coffin Planimeter 31
28. The Willis Planimeter 34

ARTICLES PAGE
29. The Roller Planimeter 35
30. Directions for Care and Use of Planimeters 36
31. Calibration of the Planimeter 37
32. Errors of Different Planimeters 40
33. Special Planimeters 42
34. Vernier Caliper 42
35. The Micrometer 43
36. The Micrometer Caliper 44
37. The Cathetometer 47
38. Aids to Computation 48

CHAPTER III.

STRENGTH OF MATERIALS. — GENERAL FORMULÆ.

39. Definitions 51
40. Stress-deformation Diagrams 53
41. Notation 55
42. Formulæ for Tension Loading 55
43. Formulæ for Compression Loading 56
44. Formulæ for Transverse Loading 57
45. Formulæ for Direct Shear 61
46. Formulæ for Torsion 61
47. Secondary Stresses 63

CHAPTER IV.

STRENGTH OF MATERIALS. — TESTING-MACHINES.

48. Testing-machines and Methods of Testing 67
49. General Character of Testing-machines 68
50. Weighing System 74
51. Frame 75
52. Power System 76
53. Shackles 76
54. Specifications for Government Testing-machine 78
55. Riehlé Brothers' Hydraulic Testing-machines 82
56. Riehlé Power Machines 84
57. Olsen Testing-machine 86
58. Thurston's Torsion Testing-machine 86
59. Power Torsion-machines 90
60. Impact-testing Machine 94
61. Machines for Testing Cement 96
62. General Requirements of Instruments for Measuring Strains 99
63. Various Forms of Extensometers 100
64. Combined Extensometer and Autographic Apparatus 107
65. Instruments for Measuring Torsion, Deflection, and Compression 111

CHAPTER V.

STRENGTH OF MATERIALS. — METHODS OF TESTING.

ARTICLES PAGE
66. Standard Methods 115
67. Tension Testing 115
68. Test-pieces of Special Materials 117
69. Directions for Tension-tests 118
70. Compression Testing 125
71. Directions for Compression-tests 125
72. Transverse Testing 126
73. Directions for Transverse Tests 127
74. Report of Transverse Tests 127
75. Torsion Testing 129
76. Directions for Torsion-tests 129
77. Testing in Direct Shear 132
78. Impact Testing 132
79. Drop-tests 133
80. Minor Tests 134
81. Special Tests and Specifications 138
82. Tests of Building Stone and Brick 152
83. Tests of Paving Material, Stones, and Ballast, Natural and Artificial 155
84. Testing Cements and Mortars 156

CHAPTER VI.

MEASUREMENT OF PRESSURE.

85. Pressure, Definitions and Units 167
86. Measurement of Pressure 168
87. Manometers 168
88. U-shaped Manometer 169
89. Cistern-manometer 171
90. Mercury Columns 172
91. Multiplying and Differential Manometers 174
92. Pressure-gauges 181
93. Vacuum-gauges 185
94. Recording-gauges 185
95. Gauge Calibration 188
96. Correction of Gauges 192
97. Forms for Calibration of Gauges 192

CHAPTER VII.

MEASUREMENT OF TEMPERATURE.

98. Temperature 193
99. Thermometers and Thermometric Materials 193
100. Thermometric Scales and Thermometric Standards 195
101. Gas Thermometers 196
102. Thermometers Employing Liquids 201

ARTICLES PAGE
103. Thermometers Employing Solids 205
104. Calorimetric Pyrometers 206
105. Electrical Methods of Measuring Temperature 209
106. Optical and Radiation Pyrometers 216
107. Calibration of Thermometers and Pyrometers 219

CHAPTER VIII.

THE MEASUREMENT OF SPEED.

108. Counting Devices 227
109. Speed Indicators Tachometers 229
110. The Chronograph 232
111. Measurement of Speed Variation within a Revolution 235

CHAPTER IX.

FRICTION. — TESTING OF LUBRICANTS.

112. Friction 236
113. Classification 237
114. Formulæ and Notation 237
115. Friction of Pivots 238
116. Friction of Cords and Belts, Belt Drive 238
117. Friction of Fluids 241
118. Lubricated Surfaces 242
119 Determinations Required 243
120 Adulteration of and Impurities in Oils 243
121. Acid Tests 245
122. Gumming or Drying 245
123. Density of Oils 246
124. Method of Finding Density 247
125. Viscosity 248
126. Method of Measuring Viscosity with Olsen Apparatus 252
127. Chill-point 252
128. Method of Finding the Chill-point 254
129. The Flash-point 254
130. Method of Testing for Flash-point 255
131. Burning-point 256
132. Method of Testing for Burning-point 256
133. Volatility 256
134. Coefficient of Friction. — Oil-testing Machines 256
135. Directions for Obtaining Coefficient of Friction with Thurston's Oil-testing Machines 264
136. Instructions for Use of Thurston's R. R. Lubricant-tester 266
137. Durability of Lubricants 266
138. Durability-testing Machine 267
139. Directions for Durability Test of Oils with Boult's Oil-testing Machine 270
140. Friction of Ball- and Roller-bearings 271
141. Forms for Report 273

CHAPTER X.

MEASUREMENT AND TRANSMISSION OF POWER.

ARTICLES PAGE
142. Definitions 275
143. Measurement of Power 276
144. Absorption Dynamometers 276
145. The Design of Prony Brakes 280
146. Cooling and Lubricating a Prony Brake 283
147. Other Forms of Friction Brakes; Self-regulating Brakes 283
148. Alden Brake 284
149. Brakes Employing the Friction of Liquids 286
150. Other Forms of Brakes: Pump Brakes 288
151. Fan Brakes 289
152. Electro-magnetic Brake 290
153. Traction Dynamometers 292
154. Transmission Dynamometers, General Types 293
155. Morin's Rotation Dynamometer 293
156. Steelyard Dynamometer 296
157. Pillow-block Dynamometer 297
158. The Lewis Dynamometer 298
159. The Differential Dynamometer 300
160. Emerson's Power Scale 304
161. The Van Winkle Power Meter 305
162. Belt Dynamometers 306
163. The Durand Dynamometer 306
164. Torsion Dynamometers 309
165. Cradle Dynamometers 314
166. Electrical Measurement of Power Input and Output 315
167. Methods of Connecting Up, etc 316
168. The Testing of Belts 318
169. The Sibley College Belt-testing Machine 318
170. Methods of Making Tests and Computations 320

CHAPTER XI.

HEAT AND THE PROPERTIES OF GASES AND VAPORS.

171. Purpose of Chapter 325
172. Notation and General Definitions 325
173. The Heat Unit and the Mechanical Equivalent of Heat 325
174. Specific Heat 326
175. Boyle's Law, Charles's Law, etc 327
176. Specific Heat of Gases 328
177. Pressure and Volume Changes in Gases 329
178. Entropy of Gases 332
179. Vaporization 333
180. Superheating and Specific Heats of Vapors 335
181. Use of Steam Tables and Diagram 337
182. Entropy of Vapors 340

ARTICLES PAGE
183. The Temperature-Entropy Diagram for Steam 341
184. Pressure and Volume Changes in Vapors 343
185. Cycles and Their Efficiencies 346

CHAPTER XII.

THE MEASUREMENT OF LIQUIDS, GASES, AND VAPORS.

186. General Considerations 356
187. Classification of Available Methods 357
188. Methods Used 357
189. Sources of Error in Weighing 360
190. Submerged Orifices 361
191. Value of the Coefficient of Discharge 362
192. Method of Calibrating an Orifice 365
193. Use of Orifices for Measuring Continuous Flow 366
194. Measurement by Means of Nozzles 368
195. The Venturi Tube or Venturi Meter for Measuring Water or Other Liquid. 369
196. The Flow of Water over Weirs 372
197. Methods of Measuring the Heads over Weirs 378
198. Conditions Affecting the Accuracy of Weirs 379
199. Effect of Disturbing Causes and Error in Weir Measurements 380
200. Directions for Calibrating Weirs, Nozzles, and Venturi Meter 380
201. Calibrated Tanks 383
202. Tank Meters 383
203. Water Meters 385
204. Hot-water Meters 394
205. The Calibration of Water Meters 396
207. Measurement of Average Velocity in Open Channels and Streams 397
208. Floats 397
209. Hydro-dynamometers and Hydrometric Pendulums 400
210. Tachometers and Current Meters 402
211. Calibration of Current Meters and Methods of Determining Average Velocity in Cross Sections 404
212. The Pitot Tube 405
213. Measurement of Average Velocity in Closed Pipes and Conduits 406
214. Conditions Affecting the Value of the Constant C in the Best Form of Pitot Tube 407
215. Calibration of Pitot Tubes 409
216. Method of Making a Traverse and Obtaining the Average or Mean Velocity 411
217. Relation between Center Velocity and Mean Velocity 412
218. The Flow of Water in Pipes 412
219. Measurement of Gases 417
220. Theoretical Velocity and Weight Discharged 419
221. Value of the Pressure P_2 421
222. Formulas for the Flow of Air 422
223. Value of the Coefficient of Discharge C 424
224. Necessary Precautions in Orifice Methods 429

ARTICLES PAGE
225 The Gasometer 430
226. Gas Meters 432
227. Dry Meters of Large Capacity 435
228. Calibration of Meters and the Relative Accuracy of Wet and Dry Meters 437
229. Gas-meter Capacities 437
230. Reservoir Method of Measuring Air or Other Gases 437
231. Anemometers 439
232. Precautions Necessary in the Use of Anemometers 441
233. Measurement of Gas by Means of Pitot Tubes 441
234. Measurement of Gas by Venturi Meter 442
235. Measurement of Gas by Calorimetry 443
236. Measurement of Vapors 445
237. Determination of Weight of Liquid 445
238. Measurement of Vapors by Means of Orifices. General Case 446
239. Method of Determining Theoretical Velocity for Steam from the Heat Chart 447
240. Value of the Pressure P_2 448
241. Weight of Vapor Discharged 449
242. Value of the Coefficient of Discharge C 449
243. Approximate Formulæ for the Flow of Steam through Orifices 452
244. Measurement of Vapors by determining Average Velocity 453
245. Steam Meters 453
246. Calorimetric Measurement of Vapors 463
247. The Flow of Steam in Pipes 463

CHAPTER XIII.

COMBUSTION AND FUELS.

248. Combustion 465
249. Weight and Volume Relations 465
250. Heat of Combustion or Calorific Power 466
251. Combustion of Elementary Fuels 466
252. Combustion of Hydrogen or of Fuels Containing Hydrogen. Higher and Lower Heating Value of Fuels 467
253. Temperatures Obtained in Combustion 468
254. Calorimetry 470
255. The Bomb Calorimeter 471
256. Method of Operating a Bomb Calorimeter 478
257. Favre and Silbermann Fuel Calorimeter 483
258. Carpenter Calorimeter 485
259. Calculations with the Carpenter Calorimeter 489
260. The Parr Calorimeter 490
261. Continuous Calorimeters 494
262. Composition of Fuels 498
263. Coal Analysis 501
264. Sampling 501
265. Moisture 502
266. Volatile Matter 502

ARTICLES PAGE
267. Determination of Fixed Carbon 504
268. The Ash Determination 505
269. Determination of Heating Value 505
270. Chemical Analysis 507
271. Sulphur Determination 509
272. Methods of Computing the Heating Value of Fuels 510
273. Buying Coal by Analysis 512
274. Flue-gas Analysis and Combustion Calculations 512
275. Sampling 513
276. Analysis of Flue-gas 517
277. Absorbing Power of the Reagents 520
278. General Forms of Flue-gas Analysis Apparatus and Method of Operation 520
279. Elliot's Apparatus 522
280. The Orsat Apparatus 524
281. The Hempel Apparatus for Gas Analysis 525
282. Automatic Flue Gas Analysis Apparatus: CO_2 Recorders 527
282. Assumptions and Accuracy of Flue Gas Analysis 530
284. Losses in Commercial Combustion The Computation of the Flue Loss 531
285. Errors in the Flue-gas Calculations as Above Carried Out 538
286. The Excess or Dilution Coefficient 538
287. Smoke Determinations 540

CHAPTER XIV.

METHODS OF DETERMINING THE AMOUNT OF MOISTURE IN STEAM.

288. Meaning of the Term "Quality of Steam" 543
289. Importance of Quality Determinations 544
290. Methods of Determining the Quality 544
291. Classification of Calorimeters 545
292. Use of Steam Tables 545
293. Method of Obtaining a Sample of Steam 546
294. Method of Inserting Thermometers 547
295. Condensing Calorimeters 548
296. Determination of the Water Equivalent of the Calorimeter 549
297. Forms of Condensing Calorimeters 550
298. Directions for Use of the Barrel Calorimeter 552
299. The Continuous Jet Condensing Calorimeter 552
300. The Hoadley Calorimeter 554
301. The Barrus Continuous Calorimeter 556
302. Superheating Calorimeters 558
303. The Barrus Superheating Calorimeter 558
304. The Throttling Calorimeter 560
305. Graphical Solutions for Throttling Calorimeter Computations 563
306. Limits of the Throttling Calorimeter 564
307. Throttling Calorimeter Constructed from Pipe Fittings 565
308. The Thomas Electric Superheating Calorimeter 566

ARTICLES PAGE
309. Methods of Direct Determination of Moisture in Steam. The Separating Calorimeter 567
310. Formulæ for Use with the Separating Calorimeter 572
311. The Limits of the Separating Calorimeter 573
312. General Method of Using the Separating Calorimeter 574
313. The Chemical Calorimeter 575
314. Universal or Combination Calorimeters 577
315. Comparative Value of Calorimeters 578

CHAPTER XV.

THE ENGINE INDICATOR.

316. Uses of the Indicator 580
317. Indicated and Dynamometric Power 581
318. Early Forms of the Steam-engine Indicator 582
319. The Richards Indicator 583
320. The Thompson Indicator 584
321. The Tabor Indicator 585
322. The Crosby Indicator 586
323. Indicators with External Springs 588
324. Indicators for Ammonia Machines 592
325. Small Piston Indicators 592
326. Large and Small Indicators 593
327. The Continuous Indicator 593
328. The Errors of the Piston Indicator 594
329. Calibration of Indicator Springs 597
330. Comparison of Dead-weight and Fluid-pressure Methods of Calibration, and Hot vs Cold Calibration 605
331. Determination of Scale of Spring and the Evaluation of Diagrams Taken with Faulty Springs 608
332. Other Errors in Indicators and Methods of Determining Their Magnitude 612
333. Method of Attaching the Indicator to the Cylinder 620
334. Reducing-motions for Indicators 621
335. Reducing Wheel 630
336. The Indicator-cord and Methods of Connecting up 632
337. Electrical Attachments for Taking a Number of Cards at the Same Time 635
338. Care of the Indicator 636
339. Directions for Taking Indicator Diagrams 636
340. The Optical Indicator 637

CHAPTER XVI.

THE INDICATOR-DIAGRAM.

341. Definitions 642
342. Measurements from the Diagrams 645
343. Indicated Horse Power 647
344. Form of the Indicator-diagram 647
345. Methods of Drawing an Hyperbola 648

ARTICLES PAGE
346. Construction of Saturation and Isentropic Curves for Steam 650
347. Clearance Determined from the Diagram 652
348. Weight of Steam Accounted for by the Indicator-diagram. The Diagram Water-rate 653
349. The Diagram Water-rate for Multicylinder Engines 655
350. Approximate Formulas for the Diagram Water-rate 656
351. Re-evaporation and Cylinder Condensation 657
352. Various Forms of Actual Steam Engine Indicator-diagrams 657
353. The Combining of Diagrams for Multicylinder Engines 661
354. The Actual Entropy Diagram for a Steam Engine 670
355. Steam-chest and Steam-pipe Diagrams 675
356. Displaced Diagrams and Time Diagrams 676
357. Theoretical and Normal Gas Engine Diagrams 678
358. The Actual Entropy Diagram for a Gas or a Mixture of Gases 685

CHAPTER XVII.

THE TESTING OF STEAM BOILERS.

359. Methods of Testing Steam Boilers 691
360. Notes on the Rules 710

CHAPTER XVIII.

THE TESTING OF STEAM ENGINES, PUMPING ENGINES, AND LOCOMOTIVES.

361. The Testing of Steam Engines. General Considerations and Definitions. 725
362. Rules for Conducting Steam-engine Tests, Code of 1902 732
363. The Testing of Pumping Engines (Steam Pumps) 774
364. The Testing of Locomotives 786
365. Calorimetric Methods of Engine Testing 799
366. Valve Setting on Steam Engines 807

CHAPTER XIX.

STEAM TURBINES.

367. General Considerations 808
368. Types of Turbines 808
369. The Single-stage Impulse Turbine 810
370. The Multi-stage (Compound) Impulse Turbine 812
371. The Multi-stage Reaction Turbine 815
372. Turbines of the Pelton Type (Open-vane or Bucket Turbines) 816
373. The Testing of Steam Turbines 817
374. Computations in Connection with Steam-turbine Tests 819
375. Separation and Estimation of Steam-turbine Losses 822
376. Method of Correcting the Results of Steam-turbine Tests for Standard or Guarantee Conditions 824

CHAPTER XX.

THE TESTING OF INJECTORS.

ARTICLES PAGE
377. General Theory of the Instrument 827
378. Practical Construction and Operation 829
379. Efficiency of Injectors 833
380. Testing of Injectors 835

CHAPTER XXI.

GAS ENGINES AND GAS PRODUCERS.

381. Types of Gas Engines 841
382. Gas-engine Fuels 846
383. Producer Gas 847
384 Types of Gas Producers 850
385. Data Relating to Gases, Fuels, etc., Generally Used in Efficiency Computations 855
386. Analysis of Fuel Gases and Determination of Heating Value 868
387. Analysis of Exhaust Gases and Computation of Heat Lost in Exhaust 877
388. Experimental Determination of Exhaust Loss 883
389. Determination of the Quantity of Producer Gas Made per Pound of Fuel. 884
390. The Testing of Gas Producers and of Gas Engines 891
391. Directions for Testing a Gas Producer 891
392. Heat Balance for a Gas Producer 899
393. Computation of Gas Producer Efficiency 903
394. Testing of Gas Engines 905
395. Heat Balance for an Internal Combustion Engine 920
396. The Computation of Gas-engine Efficiencies 921

CHAPTER XXII.

HOT-AIR ENGINES.

397. Types of Engines and Methods of Operation 925
398. The Ericsson Engine 926
399. The Rider Engine 929
400. The Testing of Hot-air Engines 933
401. Computations and Report 934

CHAPTER XXIII.

AIR-COMPRESSING MACHINERY.

402. Types of Air-compressing Machinery 938
403. Piston Compressors 938
404. Rotary or Positive Blowers 942
405. Centrifugal or Volume Blowers 943
406 Turbine Compressors 945

ARTICLES PAGE
407. Injector Blowers ... 946
408. Hydraulic Compressors ... 947
409. Physical Characteristics of Air ... 949
410. Determination of Humidity ... 951
411. The Theoretical Work of Compressing Air ... 954
412. Theoretical Compressor Horse Power ... 961
413. Effect of Clearance, Volumetric Efficiency, and Slip ... 963
414. The Efficiency of Compression and the Mechanical Efficiency of the Machine ... 965
415. Multi-stage Compression ... 967
416. The Construction of the Combined Diagram for a Two-stage Compressor and the Saving Due to Cylinder Jackets and Intercooler ... 969
417. The Computation of Efficiencies in Multi-stage Compression ... 972
418. Arrangements for Testing Piston Compressors ... 973
419. Record of Data and Computations for Air-compressor Test ... 977
420. The Action of the Centrifugal Fan ... 981
421. Work Done by a Fan: Air Horse Power ... 983
422. Fan Efficiencies ... 985
423. Arrangements for Testing Fans ... 986
424. Scope of Test and the Report ... 988

CHAPTER XXIV.

MECHANICAL REFRIGERATION.

425. Theory of Mechanical Refrigeration ... 992
426. Refrigerating Agents ... 994
427. Classification of Refrigerating Machines ... 997
428. Air Machines ... 998
429. The Absorption System ... 999
430. The Vacuum Process ... 1001
431. The Vapor-compression System ... 1003
432. Ice Making ... 1008
433. Properties of Brine Used for Cooling ... 1009
434. Insulation ... 1010
435. The Ideal Vapor-compression Refrigerating Cycle ... 1012
436. The Ideal Coefficient of Performance ... 1014
437. The Actual Vapor-compression Refrigerating Cycle ... 1016
438. The Actual Coefficient of Performance and the Real Efficiency of the Actual Refrigeration Process ... 1019
439. Wet and Dry Compression ... 1021
440. The Compressor Indicator Diagram. Volumetric Efficiency ... 1021
441. The Weight of NH_3 in Circulation ... 1024
442. Capacity and Rating of Refrigerating Machines ... 1027
443. Arrangements for Testing of Vapor-compression Refrigerating Machines ... 1028
444. Computation of Results ... 1033
445. The Heat Balance for the Refrigeration Process ... 1033

CHAPTER XXV.

HYDRAULIC MACHINERY.

ARTICLES PAGE
446. Classification.......... 1037
447. The Hydraulic Ram 1038
448. Efficiency of the Hydraulic Ram.......... 1039
449. Testing of Hydraulic Rams 1040
450. The Jet Pump 1041
451. Efficiency of the Jet Pump.......... 1043
452. Water Pressure Engines 1043
453. Efficiency of Water-pressure Engines.......... 1044
454. Vertical Water Wheels 1045
455. The Overshot Water Wheel.......... 1045
456. The Breast Wheel 1047
457. The Undershot Wheel 1048
458. The Impulse or Tangential Water Wheel 1049
459. Efficiency of Impulse Wheels 1052
460. Turbines 1052
461. Action of a Stream upon a Rotating Vane. 1054
462. The Radial Outward-flow Reaction Turbine. 1056
463. Power and Efficiency of the Radial Outward-flow Reaction Turbine . . 1057
464. Radial Inward-flow, Parallel-flow, and Mixed-flow Reaction Turbine..... 1059
465. Power and Hydraulic Efficiency of Radial Inward-flow, Parallel-flow, and Mixed-flow Reaction Turbines. 1062
466. The Testing of Water Motors 1062
467. Report and Computations 1065
468. Reciprocating Pumps 1067
469. Rotary Pumps.......... 1067
470. Centrifugal and Turbine Pumps 1068
471. The Action of the Centrifugal Pump 1068
472. Theoretical Power, Manometric, Hydraulic, and Actual Efficiencies of Centrifugal Pumps 1074
473. The Testing of Rotary and Centrifugal Pumps 1077
474. Scope of Tests and Report 1079
475. The Pulsometer 1080
476. The Humphrey Internal combustion Pump 1082
477. Pumping by Compressed Air; Pneumatic Pumps, Air Lifts. 1084

APPENDIX.

Tables.......... 1087-1110

EXPERIMENTAL ENGINEERING

CHAPTER I.

INTRODUCTORY.

Reduction of Experimental Data. Graphical Representation of Experiments.

1. **Objects of Engineering Experiments.** — The object of experimental work in an engineering course of study may be stated under the following heads: firstly, to afford a practical illustration of the principles advanced in the classroom; secondly, to become familiar with the methods of testing; thirdly, to ascertain the constants and coefficients needed in engineering practice; fourthly, to obtain experience in the use of various types of engines and machines; fifthly, to ascertain the efficiency of these various engines or machines; sixthly, to deduce general laws of action of mechanical forces or resistances from the effects or results as shown in the various tests made. The especial object for which the experiment is performed should be clearly perceived in the outset, and such a method of testing should be adopted as will give the required information.

This experimental work differs from that in the physical laboratory in its subject-matter and in its application, but the methods of investigation are to a great extent similar. In performing engineering experiments one will be occupied principally in finding coefficients relating to strength of materials or efficiency of machines; these, from the very nature of the material investigated, cannot have a constant value which will be exactly repeated in each experiment, even provided no error be made. The object will then be to find average values for these coefficients, to obtain the variation in each specific test from these average values, and, if possible, to find the law and cause of such variation.

The results are usually a series of single observations on a variable quantity, and not a series of observations on a constant quantity; so that the method of finding the probable error, by the method of least squares, is not often applicable. This method of reducing and correcting observations is, however, of such value when it is applicable that it should be familiar to engineers, and should be applied whenever practicable. The fact that single observations are all that often can be secured renders it necessary in this work to take more than ordinary precautions that such observations be made correctly and with accurate instruments.

2. **Relation of Theory to Experiment.** — It will be found in general better to understand the theoretical laws, as given in text-books, relating to the material or machine under investigation, before the test is commenced; but in many cases this is not possible, and the experiment must precede a study of the theory.

It requires much skill and experience in order to deduce general laws from special investigations, and there is always reason to doubt the validity of conclusions obtained from such investigations if any circumstances are contradictory, or if any cases remain unexamined.

On the other hand, theoretical deductions or laws must be rejected as erroneous if they indicate results which are contradictory to those obtained by experiments subject to conditions applicable in both cases.

3. **The Method of Investigation** is to be considered as consisting of three steps: firstly, to standardize or calibrate the apparatus or instruments used in the test; secondly, to make the test in such a way as to obtain the desired information; thirdly, to write a report of the test, which is to include a full description of the methods of calibration and of the results, which in many cases should be expressed graphically.

The methods of standardizing or calibrating will in general consist of a comparison with standard apparatus, under conditions as nearly as possible the same as those in actual practice. These methods will be given in detail later. The manner of performing the test will depend entirely on the experiment.

The report should be written in books or on paper of a prescribed form, and should describe clearly: (1) Object of the experiment; (2) Deduction of formulæ and method of performing the experiment; (3) Description of apparatus used, with methods of calibrating; (4) Log of results, which must include all the figures taken in the various observations of the calibration as well as in the experiment. These results should be arranged, whenever possible, in tabular form; (5) Results of the experiment; these should be expressed numerically and graphically, as explained later; (6) Conclusions deduced from the experiment, and comparison of the results with those given by theory or other experiments.

4. **Classification of Experiments.** — The method of performing an experiment must depend largely on the special object of the test, which should in every case be clearly comprehended. The following subjects are considered in this treatise, under various heads: (1) The calibration of apparatus; (2) Tests of the strength of materials; (3) Measurements of liquids and gases; (4) Tests of friction and lubrication; (5) Efficiency tests, which relate to (*a*) belting and machinery of transmission, (*b*) water-wheels, pumps, and hydraulic motors, (*c*) hot-air and gas engines, (*d*) air-compressors and compressed-air machinery, (*e*) steam-engines, boilers, injectors, and direct-acting pumps.

5. **Efficiency Tests.** — Tests may be made for various objects, the most important being probably that of determining the efficiency, capacity, or strength.

The efficiency of a machine is the ratio of the useful work delivered by the machine to the whole work supplied or to the whole energy received. The limit to the efficiency of a machine is *unity*, which denotes the efficiency of a perfect machine.

The whole work performed in driving a machine is evidently equal to the useful work, plus the work lost in friction, dissipated in heat, etc. The lost work of a machine often consists of a constant part, and in addition a part bearing some definite proportion to the useful work; in some cases all the lost work is constant.

Efficiency tests are made to determine the ratio of useful work performed to total energy received, and require the determination of, first, the work or energy received by the machine; second, the useful work delivered by the machine. The friction and other lost work is the difference between the total energy supplied and the useful work delivered. In case the efficiency of the various parts of the machine is computed separately, the efficiency of the whole machine is equal to the product of the efficiencies of the various component parts which transmit energy from the driving-point to the working-point.

The work done or energy transmitted is usually expressed in foot-pounds per minute of time, or in horse-power, which is equivalent to 33,000 foot-pounds per minute, or 550 foot-pounds per second of time.

6. Classification of Experimental Errors. — In the following articles the method of reducing observations and producing equations from experimental data is briefly set forth. All experimental observations are subject to error. The theory relating to the probability of errors is treated under the head of "Least Squares," which can be fully studied in the work by Chauvenet published by Lippincott & Company, or in the work by Merriman published by John Wiley & Sons.

The errors to which all observations are subject are of two classes: *systematic* and *accidental.*

Systematic errors are those which affect the same quantities in the same way, and may be further classified as *instrumental* and *personal.* The instrumental errors are due to imperfection of the instruments employed, and are detected by comparison with standard instruments or by special methods of calibration. Personal errors are due to a peculiar habit of the observer tending to make his readings preponderate in a certain direction, and are to be ascertained by comparison of observations: first, with those taken automatically; second, with those taken by a large number of observers equally skilled; third, with those taken by an observer whose personal error is known. Systematic errors should be investigated first of all, and their effects eliminated.

Accidental errors are those whose presence cannot be foreseen nor prevented; they may be due to a multiplicity of causes, but it is found, if the number of observations be sufficiently great, that their occurrence can be predicted by the law of probability, and the probable value of these errors can be computed by the METHOD OF LEAST SQUARES.

Before making application of the Method of Least Squares, determine the value of the systematic errors, eliminate them, and apply the Method of Least Squares to the determination of accidental errors.

7. Probability of Errors. — The following propositions are regarded as axioms, and are the fundamental theorems on which the Method of Least Squares is based:

1. Small errors will be more frequent than large ones.
2. Errors of excess and deficiency (that is, results greater or less than the true value) are equally probable and will be equally numerous.
3. Large errors, beyond a certain magnitude, do not occur. That is, the probability of a very large error is zero.

From these it is seen that the probability of an error is a function of the magnitude of the error. Thus let x represent any error and y its probability, then

$$y = f(x).$$

By combination of the principles relating to the probability of any event Gauss determined that

$$y = ce^{-h^2x^2}, \qquad (1)$$

in which c and h are constants, and e the base of the Napierian system of logarithms.

8. Errors of Single Observations. — It can be shown by calculation that the most probable value of a series of observations made on the same quantity is the arithmetical mean, and if the observations were infinite in number the mean value would be the true value. The *residual* is the difference between any observation and the mean of all the observations. The *mean error* of a *single observation* is the square root of the sum of the squares of the residuals, divided by one less than the number of observations. The *probable error* is 0.6745 times the mean error. The *error of the result* is that of a single observation divided by the square root of their number.

Thus let n represent the number of observations, S the sum of the squares of the residuals; let e, e_1, e_2, etc., represent the residual, which is the difference between any observation and the mean value; let Σ denote the sum of the quantities indicated by the symbol directly following.

Then we shall have

Mean error of a single observation $\pm\sqrt{\frac{S}{n-1}}$ (2)

Probable error of a single observation $\pm\, 0.6745\sqrt{\frac{S}{n-1}}$ (3)

Mean error of the result $\pm\sqrt{\frac{S}{n(n-1)}}$ (4)

Probable error of the result $\pm\, 0.6745\sqrt{\frac{S}{n(n-1)}}$ (5)

In every case $S = \Sigma e^2$.

9. **Example.** — The following example illustrates the method of correcting observations made on a single quantity:

A great number of measurements have been made to determine the relation of the British standard yard to the meter. The British standard of length is the distance, on a bar of Bailey's bronze, between two lines drawn on plugs at the bottom of wells sunk to half the depth of the bar. The marks are one inch from each end. The measure is standard at 72° F., and is known as the Imperial Standard Yard.

The meter is the distance between the ends of a bar of platinum, the bar being at 0° C., and is known as the *Mètre des Archives*.

The following are some of these determinations. That made by Clarke in 1866 is most generally recognized as of the greatest weight.

COMPARISON OF BRITISH AND FRENCH MEASURES.

Name of Observer	Date.	Observed length of meter in inches.	Difference from the mean. Residual = e	Square of the Residuals e^2.
Kater	1821	39.37079	−0.001460	0.0000021316
Hassler.	1832	39.38103	+ 8780	0.0000770884
Clarke.	1866	39.370432	− 1818	0.0000033124
Rogers	1884	39.37015	− 2100	0.0000044100
Comstock.	1885	39.36985	−0.002400	0.0000057600
Mean value.		39 372250		0.0000907024

$$\Sigma e^2 = S = 0.0000907024, \quad n = 5, \quad n(n-1) = 20.$$

Mean error of a single observation $= \pm\sqrt{\dfrac{S}{n-1}} = 0.00476.$

Probable error of single observation $= \pm\ 0.00317.$

Mean error of mean value $= \pm\sqrt{\dfrac{S}{n(n-1)}} = 0.00213.$

Probable error of mean value $= \pm\ 0.00142.$

That is, considering the observations of equal weight, it would be an even chance whether the error of a single observation were greater or less than 0.00317 inch, and the error of the mean greater or less than 0.00142.

10. Combination of Errors. — When several quantities are involved it is often necessary to consider how the errors made upon the different quantities will affect the result.

Since the error is a small quantity with reference to the result, we can get sufficient accuracy with approximate formulæ.

Thus let X equal the calculated or observed result, F the error made in the result; let x equal one of the observed quantities, and f its error. Then will

$$F = f\frac{dX}{dx} \tag{6}$$

in which $\dfrac{dX}{dx}$ is the partial derivative of the result with respect to the quantity supposed to vary. In case of two quantities in which the errors are F, F', etc., the probable error of the result

$$= \pm\sqrt{F^2 + F'^2} \tag{7}$$

11. Deduction of Empirical Formulæ. — Observations are frequently made to determine general laws which govern phenomena, and in such cases it is important to determine what formula will express with least error the relation between the observed quantities.

These results are *empirical* so long as they express the relation between the observed quantities only; but in many cases they are applicable to all phenomena of the same class, in which case they express *engineering* or *physical laws.*

In all these cases it is important that the form of the equation be known, as will appear from the examples to be given later. The form of the equation is often known from the general physical laws applying to similar cases, or it may be determined by an inspection of the curve obtained by a graphical representation of the experiment. A very large class of phenomena may be represented by the equation

$$y = A + Bx + Cx^2 + Dx^3 +, \text{etc.} \qquad (8)$$

In case the graphical representation of the curve indicates a parabolic form, or one in which the curve approaches parallelism with the axis of X, the empirical formula will probably be of the form

$$y = A + Bx^{\frac{1}{2}} + Cx^{\frac{3}{2}} + Dx^{\frac{5}{2}} +, \text{etc.} \qquad (9)$$

In case the observations show that, with increasing values of x, y passes through repeating cycles, as in the case of a pendulum, or the backward and forward motion of an engine, the characteristic curve would be a sinuous line with repeated changes in the direction of curvature from convex to concave. The equation would be of the form

$$y = A + B_1 \sin \frac{360^\circ}{m} x + B_2 \cos \frac{360^\circ}{m} x + C_1 \sin \frac{360^\circ}{m} 2x$$
$$+ C_2 \cos \frac{360^\circ}{m} 2x +, \text{etc.} \qquad (10)$$

Still another form which is occasionally used is

$$y = A + B \sin mx + C \sin^2 mx +, \text{etc.} \qquad (11)$$

12. Rules and Formulæ for Approximate Calculation. — A mathematical expression can often be simplified with sufficient accuracy by the omission of terms containing factors which are negligibly small as compared with the terms retained.

On the principle that the higher powers of very small quantities may be neglected with reference to the numbers themselves, we can form a series by expansion by the binomial formula, or by division, in which, if we neglect the higher powers of the smaller quantities, the resulting formulæ become much more simple, and are usually of sufficient accuracy.

Thus, for instance, let δ equal a very small fraction; then the expression

$$(a + \delta)^m = a^m + ma^{m-1}\ \delta + m\,\frac{(m - 1)}{2}\,a^{m-2}\ \delta^2 +, \text{etc.},$$

will become $a^m + ma^{m-1}\,\delta$, if the higher powers of δ be neglected. If δ is equal to $\frac{1}{1000}$ part of a, the error which results from omitting the remaining terms of the series becomes very small, as in this case the value of $\delta^2 = \frac{1}{1000000}\,a$.

The following table of approximate formulæ presents several cases which can often be applied with the effect of materially reducing the work of computation without any sensible effect on the accuracy:

$$(1 + \delta)^m = 1 + m\delta, \qquad (1 - \delta)^m = 1 - m\delta; \tag{12}$$

$$(1 + \delta)^2 = 1 + 2\,\delta, \qquad (1 - \delta)^2 = 1 - 2\,\delta; \tag{13}$$

$$\sqrt{1 + \delta} = 1 + \tfrac{1}{2}\,\delta, \qquad \sqrt{1 - \delta} = 1 - \tfrac{1}{2}\,\delta; \tag{14}$$

$$(1 + \delta)^3 = 1 + 3\,\delta, \qquad (1 - \delta)^3 = 1 - 3\,\delta; \tag{15}$$

$$\frac{1}{1 + \delta} = 1 - \delta, \qquad \frac{1}{1 - \delta} = 1 + \delta; \tag{16}$$

$$\frac{1}{(1 + \delta)^2} = 1 - 2\,\delta, \qquad \frac{1}{(1 - \delta)^2} = 1 + 2\,\delta; \tag{17}$$

$$\frac{1}{\sqrt{1 + \delta}} = 1 - \tfrac{1}{2}\,\delta, \qquad \frac{1}{\sqrt{1 - \delta}} = 1 + \tfrac{1}{2}\,\delta; \tag{18}$$

$$(1 + \delta)\,(1 + \varepsilon)\,(1 + \zeta) = 1 + \delta + \varepsilon + \zeta; \tag{19}$$

$$(1 - \delta)\,(1 - \varepsilon)\,(1 - \zeta) = 1 - \delta - \varepsilon - \zeta; \tag{20}$$

$$(1 \pm \delta)\,(1 \pm \varepsilon)\,(1 \pm \zeta) = 1 \pm \delta \pm \varepsilon \pm \zeta; \tag{21}$$

$$\frac{(1 \pm \delta)\,(1 \pm \zeta)}{(1 \pm \varepsilon)\,(1 \pm \eta)} = 1 \pm \delta \pm \zeta \mp \varepsilon \mp \eta; \tag{22}$$

$$\sqrt{pn} = \frac{p + n}{2}; \tag{23}$$

$$\sin\,(x + \delta) = \sin x + \delta \cos x; \tag{24}$$

$$\cos\,(x + \delta) = \cos x - \delta \sin x; \tag{25}$$

$$\tan\,(x + \delta) = \tan x + \frac{\delta}{\cos^2 x} = \tan x + \delta \sec^2 x; \tag{26}$$

$$\sin\,(x - \delta) = \sin x - \delta \cos x; \tag{27}$$

$$\cos\,(x - \delta) = \cos x + \delta \sin x \tag{28}$$

13. **The Rejection of Doubtful Observations.*** — It often happens that in a set of observations there are certain values which are so much at variance with the majority that the observer rejects them in adjusting the results. This might be done by application of Rule 3, Article 7, provided the magnitude of the errors which could not occur were definitely determined; but to reject such observations without proper rules is a dangerous practice, and not to be recommended.

This brings into sight a class of errors which we may term *mistakes*, and which are in no sense errors of observation, such as we have been considering. Mistakes may result from various causes, as a misunderstanding of the readings, or from recording the wrong numbers, inverting the numbers, etc.; and when it is certainly shown that a mistake has occurred, if it cannot be corrected with certainty, the observations should be rejected. *After making allowance for all constant errors, no results except those which are unquestionably mistakes should be rejected.*

The remaining discrepancies will then fall under the head of *irregular* or *accidental errors*, and are to be corrected as explained in the preceding articles; the effect of a large error is largely or wholly compensated for by the greater frequency of the smaller errors.

14. **When to Neglect Errors.** — Nearly all the observations taken on any experimental work are combined with observations of some other quantity in order to obtain the desired result. Thus, for example, in the test of a steam-engine, observations of the number of revolutions and of the mean effective pressure acting on the piston are combined with the constants giving the length of stroke and area of piston. The product of these various quantities gives the work done per unit of time.

All of these quantities are subject to correction, and it is often important to allow for such correction in the result. Just how important these corrections may be depends on the degree of accuracy which is sought.

As the degree of accuracy increases, the number of influencing circumstances increases as well as the difficulty of eliminating them;

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* See Adjustment of Observations, by T. W. Wright. N. Y., D. Van Nostrand.

hence this part of the work is often the most difficult and sometimes the most important. To what limit these corrections may be carried depends on our knowledge of the laws which govern the experiments in question, as well as the accuracy with which the observations may be taken. It is evidently unnecessary to correct by abstruse and difficult calculation for influences which make less difference than the least possible unit to be determined by observation, and this consideration should no doubt determine whether or not corrections should be taken into account or neglected.

Thus, in the case of the test of a steam-engine, we have errors made in obtaining the engine constants, i.e., length of stroke and area of piston. These errors may be simply of measurement, or they may be due to changes in the temperature of the body measured. The errors of measurement depend on accuracy of the scale used, care with which the observations are made, and can be discussed as direct observations on a single quantity. The errors due to change of temperature can be calculated if observations showing the temperature are taken and if the coefficient of expansion is known. A calculation will, in case of the steam-engine constants referred to above, show that in general the probable error of observation is many times in excess of any change due to expansion, and hence the latter may be neglected. The question of the combination of errors has already been discussed in Article 10.

It is to be remembered that the methods of correction outlined in the Method of Least Squares applies only to those accidental and irregular errors which cannot be directly accounted for by any imperfection in instruments or peculiar habit of the observer; usually the correction for instrumental and personal errors is to be made to the observations themselves, before computing the probable error.

15. Accuracy of Numerical Calculations. — The results of all experiments are expressed in figures which show at best only an approximation to the truth, and this accuracy of expression is increased by extending the number of decimal figures. It is, however, evidently true that the mere statement of an experiment, with the results expressed in figures of many decimal places, does not of necessity indicate accurate or reliable experiments. The accuracy depends not on the number of decimal places in the result, but on the least errors made in the observations themselves.

It is generally well to keep to the rule that the result is to be brought out to one more place than the errors of observation would indicate as accurate: that is, the last decimal place should make no pretensions of accuracy; the one preceding should be pretty nearly accurate. In doubtful cases have one place too many rather than too few. No mistake, however, should be made in the numerical calculations; and these, to insure accuracy, should be carried for one place more than is to be given in the result, otherwise an error may be made that will affect the last figure in the result. The extra place is discarded if less than 5; but if 5 or more it is considered as 10, and the extra place but one increased by 1.

In performing numerical calculations, it will be entirely unnecessary to attempt greater accuracy of computation than can be carried out by a four-place table of logarithms, except in cases where the units of measurement are very small and the numbers correspondingly great. In general, sufficient accuracy can be secured by the use of the pocket slide-rule, the readings of which are hardly as accurate as a three-place table of logarithms. The slide-rule will be found of great convenience in facilitating numerical computations, and its use is earnestly advised.

16. Methods of Representing Experiments Graphically. — Nearly all experiments are undertaken for the purpose of ascertaining the relation that one variable condition bears to another, or to the result. All such experiments can be represented graphically by using paper divided into squares. The result of the experiment is represented by a curve, drawn as follows: Lay off in a horizontal direction, using one or more squares as a scale, distances corresponding with the record values of one of the various observations, and in a similar manner, using any convenient scale, lay off, in a vertical direction from the points already fixed, distances proportional to the results obtained. A line connecting these various points often will be more or less irregular, but will represent by its direction the relation of the results to any one class or set of observations. A connecting line may form a smooth curve, but if, as is usually the case, the line is irregular and broken, a smooth curve should be drawn in a position representing the average value of the observations. The points of observation, located on the squared paper as described, should be distinctly marked by a cross, or a point surrounded with a circle, triangle, or square; and further, all observations of the same class should be denoted by the same mark, so that the relation of the curve to the observations can be perceived at any time.

The value of the graphical method over the numerical one depends largely on the well-known fact that the mind is more sensitive to form, as perceived by the eye, than to large numbers obtained by computation. Indeed, when numbers are used, the averages of a series of observations are all that can be considered, and the effect of a gradual change, and the relation of that change to the result, are often not perceived.

Every experiment should be expressed graphically, and students should become expert in interpreting the various curves produced. A sample of paper well suited for representing experiments is bound in the back portion of the present work. In case the horizontal distances or abscissæ represent space passed through, and the vertical distances or ordinates represent the force acting, then will the area included between this curve and the initial lines represent the product of the mean force into the space passed through, — or, in other words, the work done. The units in which the work will be expressed will depend on the scales adopted. If the unit of space represent feet, the unit of force pounds, the results will be in foot-pounds. The initial lines in each case must be drawn at distances corresponding to the scales adopted, and must represent, respectively, zero-force and zero-space.

To find the length of the mean ordinate, from which the mean pressure is easily obtained, vertical lines are drawn so close together that the portion of the curve included between them is sensibly straight; the sum of these lines, which may be expeditiously taken by transferring them successively to a strip of paper and measuring the total length, is found; and this result divided by the number gives the length of the mean ordinate. This length multiplied by the scale gives the pressure. An integrating instrument, the planimeter, is more frequently used for this purpose, and gives more accurate results. The theory of the instrument and the method of using are of great importance to engineers, and are given in full in the following chapter.

Logarithmic Cross-section Paper is very convenient for the reduction of certain forms of curves to algebraic or analytic equations. The rulings of this paper are made at distances proportional to the logarithms of the numbers which represent the ordinates and abscissæ. Any curve which may be represented by a simple logarithmic or exponential equation would be represented on paper ruled in this way by a straight line. Thus, an equation of the general form $y = Bx^n$ can be reduced so that $\log y = \log B + n \log x$, which is the equation of a straight line in logarithmic units. In this equation n is the tangent of the angle which the line makes with the axis of abscissæ, and B is the intercept on this axis from the origin. Paper ruled in this manner can be obtained from most dealers in technical supplies. In case it cannot be obtained, ordinary cross-section paper, as shown in the Appendix to this book, may be used by numbering the graduations on the axes of abscissæ and ordinates as proportional to the logarithms of the distances from the origin.

17. Autographic Diagrams. — In various instruments used in testing, a diagram is drawn automatically, in which the abscissa corresponds to the space passed through, the ordinate to the force exerted, and the area to the work done. A familiar illustration is the steam-engine indicator-diagram, in which horizontal distance corresponds to the stroke of the piston of the engine, and vertical distance or ordinates to the pressure acting on the piston at any point. The absolute amount of the pressures may be determined by reference to the atmospheric line. The distance vertically between the lines drawn on the forward and back strokes of the engine is the effective pressure acting on the piston at the given position of its stroke; the mean length of all such lines is the mean effective pressure utilized in work. The vertical distance from any point on the atmospheric line to the curve drawn while the piston is on its forward stroke is the forward pressure, the corresponding distance to the back-pressure line is the back pressure, and the areas between these respective curves give effective or total work per revolution.

An autographic device is put on many materials testing-machines: in this case the ordinates of the diagram drawn represent pressure applied to the test specimen, and abscissæ represent the stretch of the specimen. This latter corresponds to the space passed through by the force, so that the area of the diagram included between the curve and line of no pressure represents the work done, — at least so far as the resistance of the test-piece is equal to the pull exerted, which is the case within the elastic limit only.

Various dynamometers construct autographic diagrams in which ordinates are proportional to the force exerted and abscissæ to the space passed through, so that the area is proportional to the work done. The diagram so drawn would represent the work done equally well were ordinates proportional to space passed through and abscissæ to the force exerted, but such diagrams are not often used.

CHAPTER II.

APPARATUS FOR REDUCTION OF EXPERIMENTAL DATA AND FOR ACCURATE MEASUREMENT.

18. The Slide-rule. — The slide-rule is made in several forms, but it consists in every case of a sliding scale in which the distance between the divisions, instead of corresponding to the numbers marked on the scale, corresponds to the logarithms of these numbers. This scale can be made to slide past another logarithmic scale, so that by placing them in proper positions there may be shown the sum or difference of these scales, and the number corresponding. As these scales are logarithmic, the number corresponding to the sum is the product, that corresponding to the difference is the quotient. Operations involving involution and evolution can also be performed. Scales showing the logarithmic functions of angles are also usually supplied.

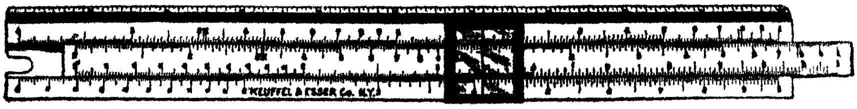

Fig. 1. — The Slide-rule.

The usual form of the slide-rule is shown in Fig. 1. This form carries four logarithmic scales, one on either edge of the slide, and one above and one below. Either scale can be used; that above is generally equivalent to one half the scale of the lower, and while not quite so accurate, is more convenient than the one below. The trigonometrical scales are on the back of the slide. The principal use of the computer is the solution of problems in multiplication and division.

The following directions for use of the plain slide-rule, which is ordinarily employed, give a simple practical method of multiplying or dividing by the slide-rule, experience having shown that when these processes are fully understood the others are mastered without instruction.

Suppose that a student has a slide-rule of the straight kind, and similar to the one in Fig. 1, which consists of a stationary scale, a sliding-scale, and a sliding pointer or runner. These parts we will term, respectively, the scale, the slide, and the runner.

19. Directions for using the Slide-rule. — Holding the rule so that the figures are right side up, four graduated edges will be seen, of which only the upper two are used in the problem we are about to describe. (The method of using the two lower scales would be exactly the same, the difference being, that they are twice as long, and that the slide is above instead of below the scale.)

Move the slide to such a position that the graduations agree throughout the length of the scale, and place the runner at a division marked 1, and the rule is ready for use. Arrange the factors to be dealt with in the form of a fraction, with one more factor in numerator than in denominator, units being introduced if necessary to make up deficiencies in the factors.

Thus, to multiply 6 by 7 by 3 and divide by 8 times 2, arrange the factors as follows:

$$\frac{6 \times 7 \times 3}{8 \times 2}.$$

The factors in the numerator show the successive positions which the runner must take; those in the denominator the positions of the slide. Thus, to solve above example, start (1) with runner at 6 on the scale, always reading from same side of runner; (2) bring figure 8 on slide to runner; (3) move runner to 7 on slide: the result can now be read on the scale; (4) bring 2 on slide to runner; (5) move runner to 3 on slide. The result is read directly on the scale at position of runner.

Another example: Multiply 11 by 6 by 7 by 8, and divide by 31. In this case arrange the factors

$$\frac{11 \times 6 \times 7 \times 8}{31 \times 1 \times 1}.$$

Start with runner at 11 on scale, move 1 on slide to runner, move runner to 6 on slide, move 1 on slide to runner, runner to 7 on slide, move 31 on slide to runner, runner to 8 on slide: read result on scale at runner.

The numbers on the slide-rule are to be considered significant figures, and to be used without regard to the decimal point. Thus the number on the rule for 8 is to be used as .8 or 80 or 800, as may be desired, even in the same problem. The significant figures in the result are readily determined by a rough computation. In case the slide projects so much beyond the scale, that the runner cannot be set at the required figure on the slide, bring the runner to 1 on the slide, then move the slide its full length, until the other 1 comes under the runner. Then proceed according to directions above; i.e., move runner to number on slide, and read results on the scale:

$$\frac{6 \times 25 \times 3.5 \times 7 \times 7 \times 31}{\pi \times 426 \times 914 \times 1 \times 1} = ?$$

Begin with the first factor in the numerator, and multiply and divide alternately, —

$$6, \quad \div \pi, \quad \times 25, \quad \div 426, \quad \times 3.5, \quad \div 914, \quad \text{etc.,}—$$

until all the factors have been used, checking them off as they are used, to guard against skipping any or using one twice. To multiply, move the runner; to divide, move the slide: in either case see that the runner points to a graduation on the slide corresponding to the factor. The result at the end or at any stage of the process is given by the runner on the stationary scale. Or, to be more exact, the significant figures of the result are given, for in no case does the slide-rule show where to place the decimal point. If the decimal point cannot be located by inspection of the factors, make a rough cancellation.

Involution and evolution are readily mastered by simple practice. Slide-rules working on the same principle are frequently made with circular or cylindrical scales, which in the Thatcher and Fuller instruments are of great length.

Thatcher's calculating instrument consists of a cylinder 4 inches in diameter and 18 inches long, working within a framework of triangular bars. Both the cylinders and bars are graduated with a double set of logarithmic scales, and results in multiplication or division can be obtained from one setting of the instrument, hence it is especially convenient when a series of numbers are to be multiplied by a common factor. The scales in this instrument are about 50 feet in length, and results can be read usually to five places.

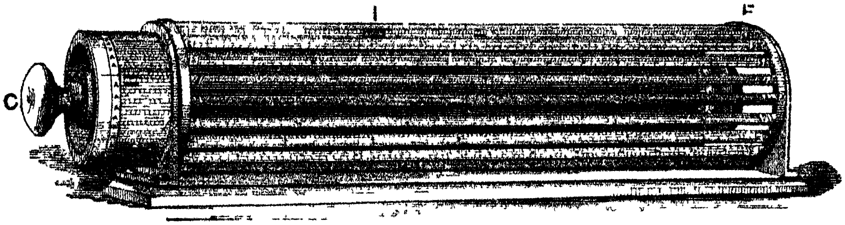

FIG. 2. — THE THATCHER RULE.

The instrument is similar to the straight slide-rule previously described, the scale on the triangular bars corresponding to the stationary scale, that on the cylinder to the sliding scale, and a triangular index *I* to the sliding pointer or runner. The method of using is essentially similar to that of the plain slide-rule, thus, to solve an example of the form a/b, put the runner *I* on the triangular scale at the number corresponding to a, bring the number corresponding to b on the cylindrical scale to register with a on the triangular scale; the respective numbers on the triangular scale and cylinder will in this position all be in the ratio of a to b, and the quotient will be read by noting that number on the triangular scale which registers with 1 on the cylindrical scale. The product of this quotient by any other number will be obtained by reading the number on the triangular scale registering with the required multiplier on the cylindrical scale.

Fuller's slide-rule, Fig. 3, consists of a cylinder *C* which can be moved up or down and turned around a sleeve which is attached to the handle *H*. A single logarithmic scale, 42 feet in length, is graduated around the cylinder spirally, and the readings are obtained by means of two pointers or indices, one of which, *A*, is attached to the handle, and the other, *B*, to an axis which slides in the sleeve. This instrument is not well adapted for multiplying or dividing a series of numbers by a constant, since the cylinder must be moved for every result. The instrument is, however, very convenient for ordinary mathematical computations, and the results may be read accurately to four decimal places.

The method of using the instrument is as follows: Call the pointer A, fixed to the handle, the *fixed pointer*, the other, BB', which may be moved independently, the *movable index*. To use the instrument, as for example in performing the operation indicated by $(a \times b) \div c$, set the fixed pointer A to the first number in the numerator, then bring the movable index B to the first figure in the denominator; then move the cylinder C until the second figure in the numerator appears under the movable index; finally read the answer on the cylinder C underneath the fixed pointer A.

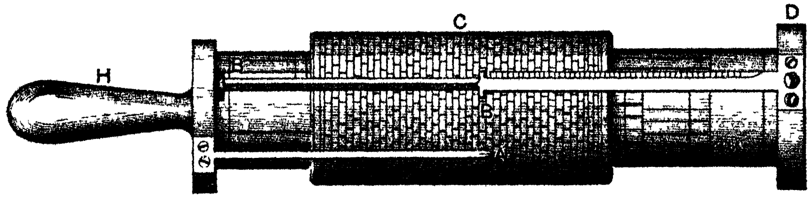

FIG. 3. — THE FULLER SLIDE-RULE.

In general, to divide with this instrument move the index B; to multiply, move the cylinder C; read results under the fixed pointer A. The movable index BB' has two marks, one at the middle, the other near the end of the pointer, either of which may be used for reading, as convenient, their distance apart corresponding to the entire length of the scale on the cylinder C.

20. The Vernier. — The *vernier* is used to obtain finer subdivisions than is possible by directly dividing the main scale, which in this discussion we will term the *limb*.

The vernier is a scale which may be moved with reference to the main scale or limb or, *vice versa*, the vernier is fixed and the limb made to move past it.

The vernier has usually one more subdivision for the same distance than the limb, but it may have one less. The theory of the vernier is readily perceived by the following discussion. Let d equal the value of the least subdivision of the limb; let n equal the number of subdivisions of the vernier which are equal to $n - 1$ on the limb. Then the value of one subdivision on the vernier is

$$d\left(\frac{n-1}{n}\right).$$

The difference in length of one subdivision on the limb and one on the vernier is

$$d - d\left(\frac{n-1}{n}\right) = \frac{d}{n},$$

which evidently will equal the least reading of the vernier, and indicates the distance to be moved to bring the first line of the vernier to coincide with one on the limb. In case there is one more subdivision on the limb than on the vernier for the same distance, the interval between the graduations on the vernier is greater than on the limb, and the vernier must be behind its zero-point with reference to its motion, and hence is termed *retrograde.* The formula for this case, using the same notation as before, gives $d\left(\frac{n+1}{n}\right) - d = \frac{d}{n}$ for the least reading.

The following method will enable one to readily read any vernier: 1. Find the value of the least subdivision of the limb. 2. Find the number of divisions of the vernier which corresponds to a number one less or one greater than that on the limb: the quotient obtained by dividing the least subdivision of the limb by this number is the value of the least reading of the vernier. The following rules for reading should be carefully observed:

Firstly. *Read the last subdivision of the limb passed over by the zero of the vernier on the scale of the limb as the reading of the limb.*

Secondly. *Look along the vernier until a line is found which coincides with some line on the limb. Read the number of this line from the scale of the vernier. This number multiplied by the least reading of the vernier is the reading of the vernier.*

Thirdly. *The sum of these readings is the one sought.*

Thus, in Fig. 5, page 22, (1) the reading of the limb is 4.70 at *a*; (2) that of the vernier is 0.03; (3) the sum is 4.73.

21. The Polar Planimeter. — The planimeter is an instrument for evaluating the areas of irregular figures, and in some one of its numerous forms is extensively used for finding the areas of indicator and dynamometer diagrams.

The principal instrument now in use for this purpose was invented by Amsler and exhibited at the Paris Exposition in 1867. This form is now generally known as Amsler's Polar Planimeter;

as most of the other instruments are modifications of this one, it is important that it be thoroughly understood.

The general appearance of the instrument is shown in Fig. 4, from which it is seen that it consists of two simple arms *PK* and *FK*, pivoted together at the point *K*. The arm *PK* during use is free to rotate around the point *P*, and is held in place by a weight. The arm *KF* carries at one end a tracing-point, which is passed around the borders of the area to be integrated. It also carries a wheel, whose axis is in the same vertical plane with the arm *KF*, and which may be located indifferently between *K* and *F*, or in *KF* produced. It is usually located in *KF* produced, as at *D*. The rim of this wheel is in contact with the paper, and any motion of the arm, except in the direction of its axis, will cause it to revolve. A graduated scale with a vernier denotes the amount of lineal travel of its circumference. This wheel is termed the *record-wheel*, Fig. 5. The wheel *D* is subdivided into a given number of parts,

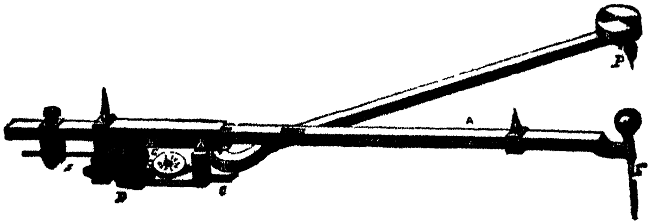

FIG. 4.—AMSLER'S POLAR PLANIMETER.

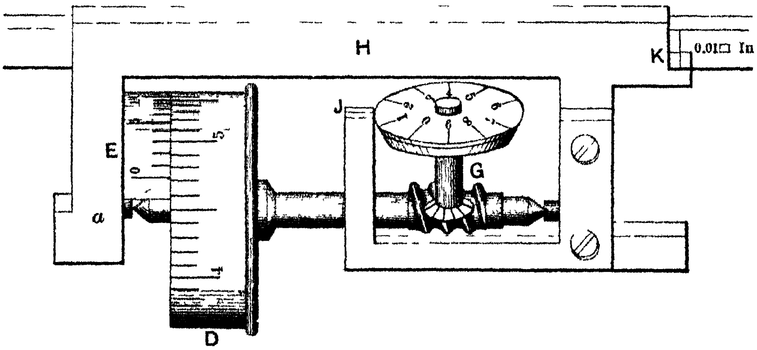

FIG. 5.—RECORD WHEEL, AMSLER POLAR PLANIMETER.

usually 100; the value of one of these parts is to be obtained by dividing the circumference of the rim of the wheel which is in contact with the paper by the number of divisions. This result will give the value of the least division on the limb; this is subdivided by an attached vernier, in this particular case to tenths of the reading of the limb, so that the least reading of the vernier is one thousandth of that of one revolution.

22. Theory of the Instrument. — There are a number of methods of developing the theory of the planimeter, two of which will be given. The first of these is due to Mr. G. B. Upton, and was published in the "Sibley Journal" for 1905-06.

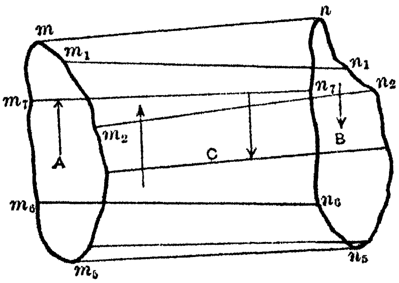

FIG. 6.

If a straight line *mn* move in a plane, it will generate an area. This area may be considered positive or negative according to the direction of motion of the line. In Fig. 6, let the paths of the ends *m* and *n* of the line be the perimeters of the areas *A* and *B* respectively, *m* and *n* travelling clockwise about these perimeters. The areas *A*, *B*, and *C* between *A* and *B*, are then generated by the line as shown by the arrows. Calling *A* a positive area, it is at once apparent that the net area generated is

$$A + C - C - B, \text{ or } A - B.$$

The immediate corollary to this is that if the area *B* be reduced in width to zero, as in Fig. 7, i.e., becomes a line along which *n* travels back and forth, the area swept over will be *A*, around which *m* is carried.

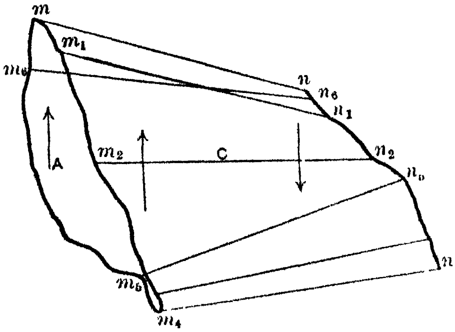

FIG. 7.

Analyzing a differential motion of the line from *mn* to *m'n'*

(Fig. 8), it may be broken up into three parts: a movement perpendicular to the line, giving area ldp; a movement in the direction of the length of the line, giving no area; and a movement of rotation about one end, giving as area $\frac{1}{2}l^2d\theta$. The total differential of area is then $dA = ldp + \frac{1}{2}l^2d\theta$. l is always a constant during the operation of a planimeter, so that

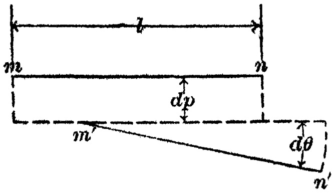

FIG. 8.

$$A = \int dA = l\int dp + \frac{1}{2}l^2\int d\theta.$$

The common use of a planimeter is that typified in Fig. 7, where the tracing-point is carried around the area to be measured, while the other end of the tracing-arm is guided back and forth along some line. The guide-line is usually either a straight line or an arc of a circle. When the tracing-point has returned to its initial position the net angle turned through by the tracing-arm, or $\int d\theta$, is zero. Hence $A = l\int dp$ simply. But $\int dp$ is the net distance the arm has moved perpendicular to itself. Call this R, and there results the equation of the planimeter

$$A = lR. \qquad (1)$$

The Zero-circle (see Fig. 9).—If the two arms be clamped so that the plane of the record-wheel intersects the centre P, and be revolved around P, the graduated circle will be continually travelling in the direction of its axis, and will evidently not revolve. A circle generated under such a condition around P as a centre is termed the zero-circle. If the instrument be unclamped and the tracing-point be moved around an area in the direction of the hands of a watch outside the zero-circle, the registering wheel will give a positive record; while if it be moved in the same direction around an area inside the zero-circle, it will give a negative record. This fact makes it necessary, in evaluating areas that are very large and have to be measured by swinging the instrument completely around P as a centre, to know the area of this zero-circle, which must be added to the determination given by the instrument, since for such cases that circumference is the initial point for measurement.

If the polar planimeter is so used as to bring in the zero-circle, the case is that of Fig. 6, each end of the line describing an area. The tracing-arm sweeps over the difference between the area described by *T* (Fig. 9) and the circle made by *G* about *P* as a centre. This difference-area is not, however, recorded by the planimeter because the $\int d\theta$ is now 2π instead of zero, *T* making a complete revolution about *P*. The linear turning of the edge of the recording-wheel is $\int dp - 2\pi n$, where n is the distance from guided point *G* to the plane of the wheel. The effect on the reading is the same as if the radius *PG* were increased. The zero-circle is traced by *T* when the plane of *W* passes through *P*. Then $\int dp = 2\pi n$, and the wheel records zero.

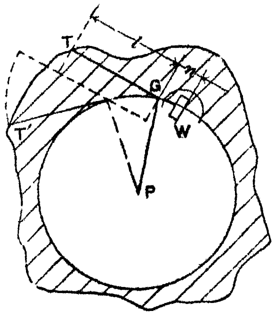

FIG. 9.

In practice the area described by the tracing-point is found by adding to the area of the zero-circle the area recorded by the wheel, taking account of the algebraic sign of the latter.

The second demonstration of the theory of the planimeter is given by Professor A. G. Greenhill, published in his "Differential and Integral Calculus," p. 228.

The planimeter in its most usual form, that invented by Amsler of Schaffhausen, consists of two bars *OA*, *AP*, jointed at *A*, and carrying in *PA* produced a small graduated roller *R*, with axis fixed parallel to *PA* (Fig. 10).

To explain the theory of the instrument, let $OA = a$, $AP = b$, $AR = c$, and the radius of the roller $= r$; and let the direction of a positive rotation of the roller, as marked by the graduations, be that of rotation on a right-handed screw on the axle of *R*, which would give a motion in the direction *AR*.

Drop the perpendicular *OI* from *O* on *AR*, and first suppose the joint *A* clamped.

Then if I is in AR produced, a rotation of the instrument about O with *angular* velocity $\frac{d\theta}{dt}$ will give to R the component velocities $OI\frac{d\theta}{dt}$ in the direction IR and $IR\frac{d\theta}{dt}$ perpendicular to IR, and therefore compel the roller to turn with angular velocity $\frac{RI}{r}\,\frac{d\theta}{dt}$; but when I is on the other side of R, the angular velocity of the roller will be $-\frac{RI}{r}\,\frac{d\theta}{dt}$.

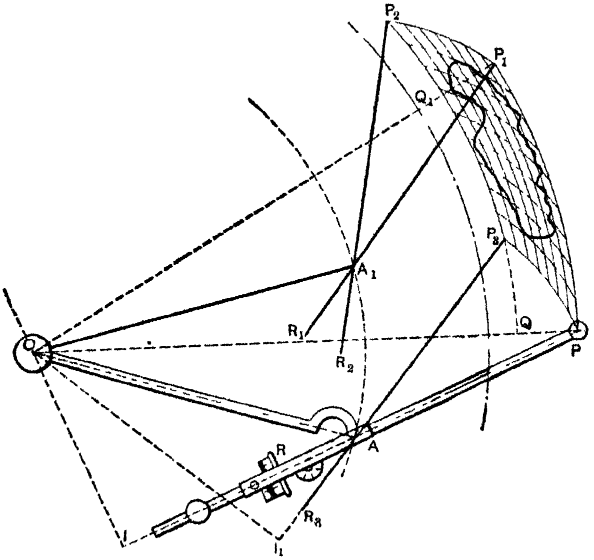

FIG 10.

Therefore, keeping A clamped, the roller will turn through an angle $\frac{RI}{r}\,\theta$ or $-\frac{RI}{r}\,\theta$, according as I is not or is on the same side of R as A, when the instrument is rotated through an angle θ about O.

When I coincides with R, the roller will not turn, and then P describes a circle called the zero-circle, represented by the middle dotted circular line (Fig. 10) of radius

$$\sqrt{(OR^2 + RP^2)} = \sqrt{\{a^2 - c^2 + (b + c)^2\}} = \sqrt{(a^2 + b^2 + 2\,bc)}.$$

Next unclamp the joint A, and clamp O; the roller will turn with angular velocity $-\frac{c}{r}\frac{d\phi}{dt}$ when the bar AP turns with angular velocity $\frac{d\phi}{dt}$, so that the roller will turn through an angle $-\frac{c}{r}\phi$ while AP turns through an angle ϕ.

Now suppose P to travel round the finite circuit $PP_1P_2P_3$ by a combination of the preceding motions in the following order:

1. Clamp the joint A, and move P to P_1, and A to A_1, on arcs of circles with center O; then the roller will turn through an angle $\frac{RI}{r}\theta$, if the angle $AOA_1 = POP_1 = \theta$.

2. Unclamp A and clamp O, and move P_1 to P_2 on the arc of a circle of center A_1; then the roller will move through an angle $-\frac{c}{r}\phi$, if the angle $P_1A_1P_2 = \phi$.

3. Unclamp O and clamp A, and move P_2 backwards to P_3 and A_1 to A on arcs of circles with center O, through an angle θ; then the roller will move through an angle $-\frac{RI_1}{r}\theta$, if OI_1 is the perpendicular from O on P_3A.

4. Unclamp A and clamp O, and move P_3 to P on the arc of a circle of center A, and consequently through an angle ϕ; the roller will turn through an angle $\frac{c}{r}\phi$, which cancels the angle due to motion (2).

In completing the finite circuit $PP_1P_2P_3$ the roller will then have turned through an angle

$$(RI - RI_1)\frac{\theta}{r}\cdot$$

But the area $PP_1P_2P_3$ = area PP_1Q_1Q

$= \text{sector } OPP_1 - OQQ_1.$

$= \tfrac{1}{2}(OP^2 - OP_3^2)\theta.$

$= \tfrac{1}{2}(OA^2 + AP^2 + 2\,AI\cdot AP - OA^2 - AP^2 - 2\,AI_1\cdot AP)\theta.$

$= (AI - AI_1)b\theta.$

$=$ br times the angle (in cm.) turned through by the roller.

The area $PP_1P_2P_3$ is therefore b times the travel of the circumference of the roller, so that by altering the length b by an adjustment on the instrument, the area can be read off in any required units.

Any irregular area must be supposed to be built up of infinitesimal elements found in the same manner as $PP_1P_2P_3$, and will be accurately measured by the roller when the point P completes a circuit of the perimeter, both joints being now free to turn simultaneously.

If, however, the origin O is inside the area, the area of the zero-circle must be added to the reading of the roller.

23. Forms of Polar Planimeters. — Polar planimeters are made in two forms: 1. With length of tracing arm PA, Fig. 10, fixed. 2. With length of tracing arm PA adjustable so that l, Eq. (1), may be varied. Since the area is in each case equal to the length of this arm multiplied by the lineal space R moved through by the record-wheel, we have in the first case, since l is not adjustable, the result always in the same unit, as square inches or square centimeters. In this case it is customary to fix the circumference of the record-wheel and compute the arm l so as to give the desired units.

For example, the circumference of the record-wheel is assumed as equal to 100 divisions, each one-fortieth of an inch, thus giving us a distance of 2.5 inches traversed in one revolution. The diameter corresponding to this circumference is 0.796 inch, which is equal to 2.025 centimeters. The distance from pivot to tracing-point can be taken any convenient distance: thus, if the diameter of the record-wheel is as above, and the length of the arm be taken as 4 inches, the area described by a single revolution of the register-wheel will be $2.5 \times 4 = 10.0$ square inches.

Since there were 100 divisions in the wheel, the value of one of these would be in this case 0.1 square inch. This would be subdivided by the attached vernier into ten parts, giving as the least reading one one-hundredth of a square inch. By making the arm larger and the wheel smaller, readings giving the same units could be obtained.

The formula expressing this reduction is as follows: Let d equal the value of one division on the record-wheel; let l equal the length of the arm from pivot to tracing-point; let A equal the area, which must evidently be either 1, 10, or 100 in order that the value of the readings in lineal measures on the record-wheel shall correspond with the results in square measures. Then by equation (1) we shall have, supposing 100 divisions,

$$100\,dl = A; \tag{2}$$

$$l = \frac{A}{100\,d}. \tag{3}$$

If $A = 10$ square inches and $d = \frac{1}{40}$ inch,

$$l = \frac{10}{2.5} = 4.$$

If $A = 10$ square inches and $d = \frac{1}{50}$ inch,

$$l = \frac{10}{2} = 5.$$

The length of the arm from center to the pivot has no effect on the result unless the instrument makes a complete revolution around the fixed point E, in which case the area of the zero-circle must be considered. It is evident, however, that this arm must be taken sufficiently long to permit free motion of the tracing-point around the area to be evaluated.

The second class of instruments, shown in Fig. 4, is arranged so that the pivot can be moved to any desired position on the tracing-arm KF, or, in other words, the length can be changed to give readings in various units. The effect of such a change will be readily understood from the preceding discussion.

24. The Mean Ordinate by the Polar Planimeter. — If we let p equal the length of the mean ordinate, and let L equal the length of the diagram, Fig. 11, then the area $A = Lp$, but the area $A = lR$ [Eq. (1), p. 24]. Therefore $Lp = lR$, from which

$$l \div L = p \div R. \tag{4}$$

In an instrument in which l is adjustable, it may be made the length of the area to be evaluated. Now if l be made equal L, $p = R$. *That is, if the adjustable arm be made equal to the length of the diagram, the mean ordinate is equal to the reading of the record-wheel, to a scale to be determined.*

The method of making the adjustable arm the length of the diagram is facilitated by placing a point *U* on the back of the planimeter at a convenient distance back of the tracing-point *F* and mounting a similar point *V* at the same distance back of the pivot *C*; then in all cases the distance *UV* will be equal to the length of the adjustable arm *l*. The instrument is readily set by loosening the set-screw *S* and sliding the frame carrying the pivot and record-wheel until the points *UV* are at the respective ends of the diagram to be traced, as shown in Fig. 11.

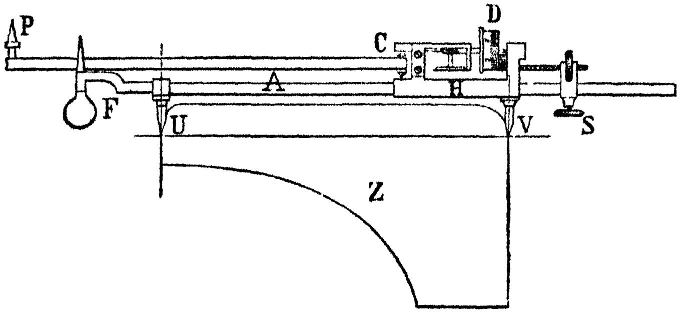

Fig. 11. — Method of Setting the Planimeter for Finding the Mean Ordinate.

In the absence of the points *U* and *V* the length of the diagram can be obtained by a pair of dividers, and the distance of the pivot *C* from the tracing-point *F* made equal to the length of the diagram. In this position, if the tracing-point be carried around the diagram, the reading will be the mean ordinate of the diagram expressed in the same units as the subdivisions of the record-wheel; thus, if the subdivisions of this wheel are fortieths of one inch, the result will be the length of the mean ordinate in fortieths. This distance, which we term the scale of the record-wheel, is not the distance between the marks on the graduated scale, but is the corresponding distance on the edge of the wheel which comes in contact with the paper.

The scale of the record-wheel evidently corresponds to a linear distance, and it should be obtained by measurement or computation. It is evidently equal to the number of divisions in the circumference divided by πd, in which d is the diameter, or it can be obtained by measuring a rectangular diagram with a length equal to l, and a mean ordinate equal to one inch, in which case the reading of the record-wheel will give the number of divisions per inch. A diameter of 0.795 inch, which corresponds to a radius of one centimeter, with a hundred subdivisions of the circumference, corresponds almost exactly to a scale of forty subdivisions to the inch, and is the dimension usually adopted on foreign-made instruments.

25. The Suspended Planimeter. — In the Amsler Suspended Planimeter, as shown in Fig. 12, pure rolling motion without slipping is assumed to take place. The motion of the record-wheel, not clearly shown in the figure, is produced by the rotation of the cylinder c in contact with the spherical segment K. The rotation of the segment is due to angular motion around the pole O, that of the cylinder c to its position with reference to the axis of the segment. This position depends on the angle that the tracing arm, ks, makes with the radial arm, BB, the area in each case being, as with the polar planimeter, equal to the product of the length of the tracing arm from pivot to tracing point multiplied by a constant factor.

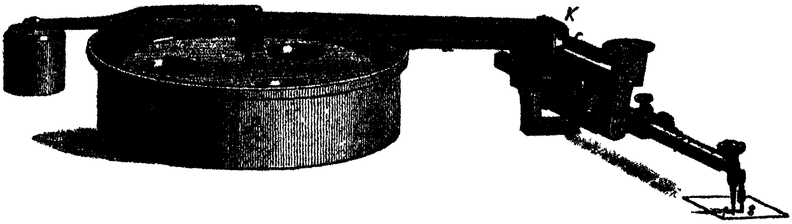

FIG. 12. — SUSPENDED PLANIMETER.

26. The Coffin Planimeter and Averaging Instrument. — The instrument is shown in Fig. 13, from which it is seen that it consists of an arm supporting a record-wheel whose axis is parallel to the line joining the extremities of the arm. This instrument was invented by the late John Coffin, of Johnstown, in 1874. The record-wheel travels over a special surface; one end of the arm travels in a slide, the other end passes around the diagram.

27. Theory of the Coffin Instrument. — This planimeter may be considered a special form of the Amsler, in which the point P, see

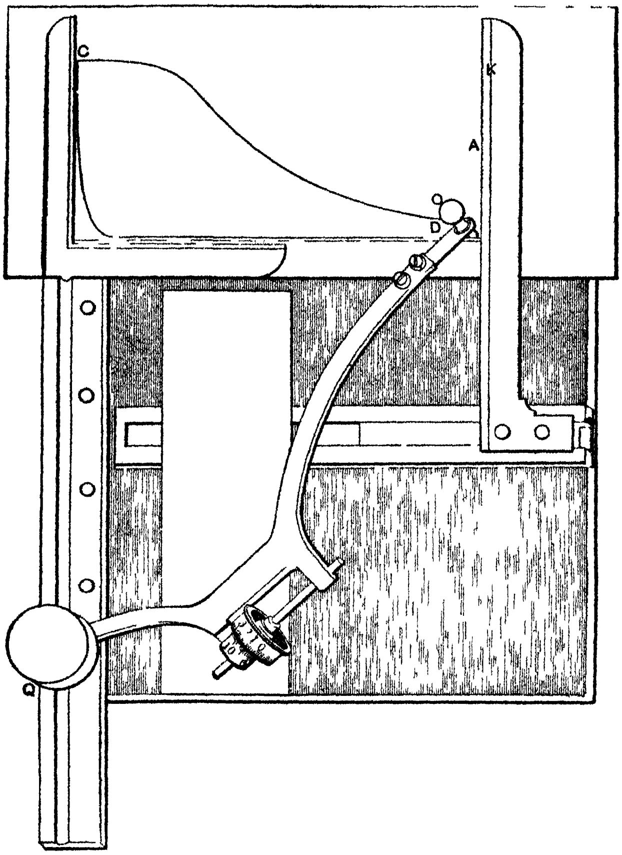

Fig. 13.—Coffin Averaging Instrument

Fig. 14, page 33, moves in a right line instead of swinging in an arc of a circle, and the angle CPT is a fixed right angle. The differential equation for area therefore is

$$dA = ln d\theta, \tag{5}$$

and the differential equation of the register becomes

$$dR = n d\theta. \tag{6}$$

Hence, as in equation (1),

$$A = lR. \tag{7}$$

That is, the area is equal to the space registered by the record-wheel multiplied by the length of the planimeter arm.

This instrument may be made to give a line equivalent to the mean ordinate (M. O.) by placing the diagram so that one edge is in line with the guide for the arm; starting at the farthest portion of the diagram, run the tracing-point around in the usual manner to the point of starting, after which run the tracing-point perpendicular to the base along a special guide provided for that purpose until the record-wheel reads as at the beginning. This latter distance is the mean ordinate.

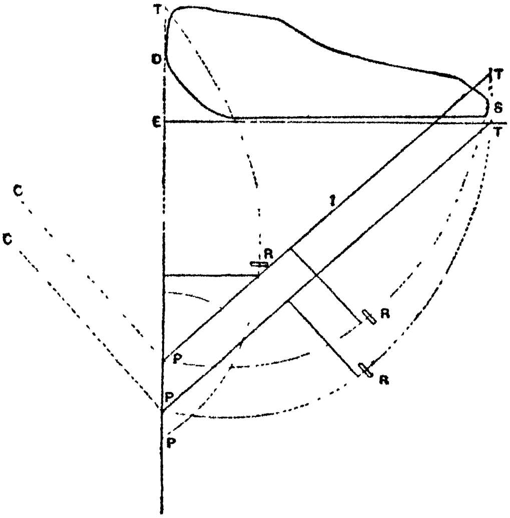

FIG. 14.—COFFIN AVERAGING INSTRUMENT.

To prove, take as in Art. 24 the M. O. = p, the length of diagram = L, the perpendicular distance = S. Then

$$A = pL = lR. \tag{8}$$

Let C be the angle, EPT, that the arm makes with the guide, Fig. 14. In moving over a vertical line this angle will remain constant, and the record will be

$$R = S \sin C. \tag{9}$$

For the position at the end of the diagram

$$\sin C = L \div l;$$

therefore

$$R = SL \div l.$$

Substituting this in equation (8),

$$pL = lR = lSL \div l = SL.$$

Hence $p = S$, which was to be proved.

From the above discussion it is evident that areas will be measured accurately in all positions, but that to get the M. O. the base of the diagram must be placed perpendicular to the guide, and with one end in line of the guide produced.

It is also to be noticed that the record-wheel may be placed in any position with reference to the arm, but that it must have its axis parallel to it, and that it registers only the perpendicular distance moved by the arm.

28. The Willis Planimeter. — This planimeter is of the same general type as the Amsler Polar, but in place of the record-wheel for recording-arm it employs a disk or sharp-edged wheel free to slide on an axis perpendicular to the tracing-arm. The distance moved perpendicular to this arm is read on the graduated edge of a triangular scale which is supported in an ingenious manner, as shown in the accompanying figure. The planimeter-arm can be adjusted as in the Amsler Planimeter so as to read the M.E.P. direct. An adjustable pin, E, is employed for the purpose of setting off the length of the diagram.

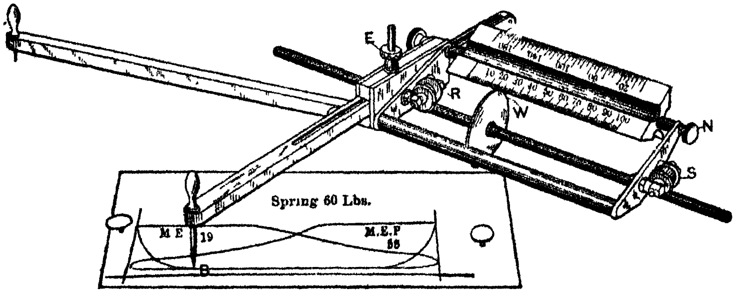

FIG. 15. — THE WILLIS PLANIMETER.

The mathematical demonstration is exactly as for the Amsler Planimeter, but in this case it is evident that the perpendicular distance which is registered on the scale is independent of the circumference of the wheel. The only conditions of accuracy are, that the axis of the scale shall be at right angles to the arm of the planimeter, and that its graduations shall be equal to the area to be measured divided by the length of the arm.

29. **The Roller Planimeter.** — This is the most accurate of the instruments for integrating plane areas, and is capable of measuring the area of a surface of indefinite length and of limited breadth. This instrument was designed by Herr Corradi of Zürich, and is manufactured in this country by Fauth & Company of Washington, D. C.

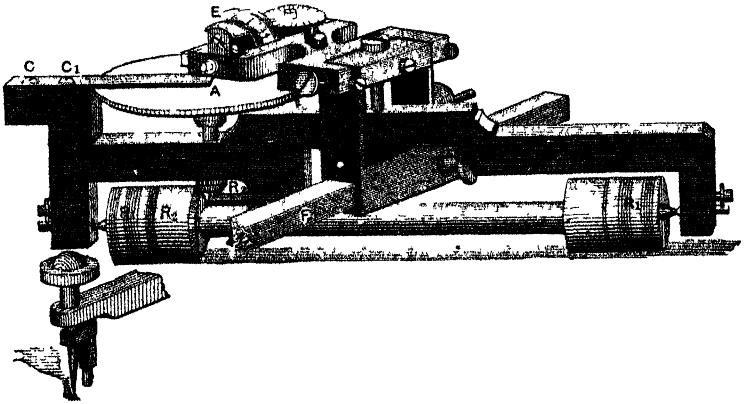

Fig. 16. — Roller Planimeter.

A view of the instrument is shown in Fig. 16. The features of this instrument are: first, the unit of the vernier is so small that surfaces of quite diminutive size may be determined with accuracy; secondly, the space that can be encompassed by one fixing of the instrument is very large; thirdly, the results need not be affected by the surface of the paper on which the diagram is drawn; and, fourthly, the arrangement of its working parts admits of its being kept in good order a long time.

The frame *B* is supported by the shaft of the two rollers R_1R_1, the surfaces of which are fluted. To the frame *B* are fitted the disk *A*, and the axis of the tracing-arm *F*. The whole apparatus is moved in a straight line to any desired length upon the two rollers resting on the paper, while the tracing-point travels around the diagram to be integrated. Upon the shaft that forms the axis of the two rollers R_1R_1 a minutely divided miter-wheel R_2 is fixed, which gears into a pinion R_3. This pinion, being fixed upon the same spindle as the disk *A*, causes the disk to revolve, and thereby actuates the entire recording apparatus.

The measuring-roller *E*, resting upon the disk *A*, travels radially thereon in accord with the motion of the tracing-arm *F*, this measuring-roller being actuated by another arm fixed at right angles to the tracing-arm and moving freely between pivots. The axis of the measuring-roller is parallel to the tracing-arm *F*. The top end of the spindle upon which the disk *A* is fixed pivots on a radial steel bar CC_1, fixed upon the frame *B*.

30. Directions for Care and Use of Planimeters. — The revolving parts should spin around easily but at the same time accurately, and the various arms should swing easily and show no lost motion. The pitch-line of the record-wheel should be as close as possible to the vernier, but yet must not touch it; the counting-wheel must work smoothly, but in no way interfere with the motion of the record-wheel. Oil occasionally with a few drops of watch or nut oil. Keep the rim of the record-wheel clean and free from rust. Wipe with a soft rag if it is touched with the fingers.

Prepare a smooth level surface, and cover it with heavy drawing-paper, for the record-wheel to move over. Stretch the diagram to be evaluated smooth.

Handle the instrument with the greatest care, as the least injury may ruin it. Select a pole-point so that the instrument will in its initial position have the tracing-arm perpendicular either to the pole-arm or to the axis of the fluted rollers, as the case may be; for in this position only is the error neutralized which arises from the fact that the tracer is not returned to its exact starting-point. Then marking some starting-point, trace the outline of the area to be measured in the direction of the hands of a watch, slowly and carefully, noting the reading of the record-wheel at the instant of starting and stopping. It is generally more accurate to note the initial reading of the record-wheel than to try to set it at zero.

Special Directions. — To obtain the mean ordinate with the polar planimeter, make the length of the adjustable arm equal to the length of the diagram, as explained in Art. 24, page 29, and follow directions for use as before.

In *using the Coffin planimeter*, the grooved metal plate is first attached to the board upon which the apparatus is mounted, as shown in the cut, page 32, being held in place by a thumb-screw applied to the back side.

The diagram will be held securely in place by the spring-clips adjacent, *A* and *C*, Fig. 13. The area may be found by running the tracing-point around the diagram, as described for the polar planimeter, for any position within the limits of the arm. The mean ordinate may be found by locating the diagram as shown in the cut, with one extreme point in the line of the metal groove produced, and the dimensions representing the length of the diagram perpendicular to this groove. Start to trace the area at the point of the diagram farthest from the metal groove produced, as shown in Fig. 13; pass around in the direction of the motion of the hands of a watch to the point of beginning; then carry the tracing-point along the straight-edge, *AK*, which is parallel to the metal groove, until the record-wheel shows the same reading as at the instant of starting: this vertical distance is the length of the mean ordinate.

31. Calibration of the Planimeter.—In order to ascertain whether the instrument is accurate and graduated correctly, it is necessary to resort to actual tests to determine the character and amount of error.

It is necessary to ascertain: 1. Whether the axis of record-wheel is parallel to axis of tracing-arm. 2. Whether the position of the tracing-arm vernier is correct, and agrees with the constants tabulated or marked on the tracing-arm.

These tests are all made by comparing the readings of the instrument with a definite and known area. To obtain a definite area, a small brass or German-silver rule, shown at *L*, Fig. 17, is used; this rule has a small needle-point near one end, and a series of small holes at exact distances of one inch or one centimeter from one another, starting at unit distance from the needle-point. To use the rule the needle-point is fixed on a smooth surface covered with paper, the planimeter is set with its tracing-point in one of the holes of the rule, and the pole-point fixed as required for actual use. With the tracing-point in the rule describe a circle, as shown by the dotted lines (Fig. 17), around the needle-point as a centre. Since the radius of this circle is known, its area is known; and as the tracing-point of the planimeter is guided in the circumference, the reading of the record-wheel should give the correct area.

The method of testing is illustrated in Figs. 17, 18, 19, and 20. Figs. 17 and 19 show the method with reference to the polar planimeter; Figs. 18 and 20 show the corresponding methods of testing the rolling planimeter. In Figs. 17 and 19, *P* is the position of the pole, *B* the pole-arm, and *A* the tracing-arm. In Figs. 18 and 20, *B* is the axis of the rollers and *A* is the tracing-arm.

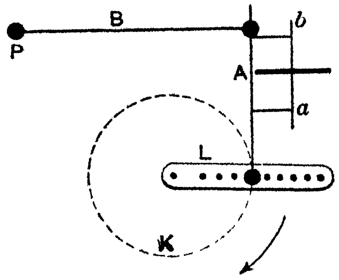

FIG. 17.

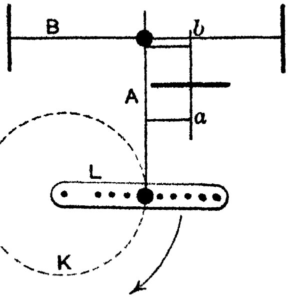

FIG. 18.

First Test. This operation, see Figs. 17 and 18, consists in locating the planimeters as shown, and then slowly and carefully revolving so as to swing the check-rule as shown by the arrow. Take readings of the vernier at initial point, and again on returning to the starting-point: the difference of these readings should give the area. Repeat this operation several times.

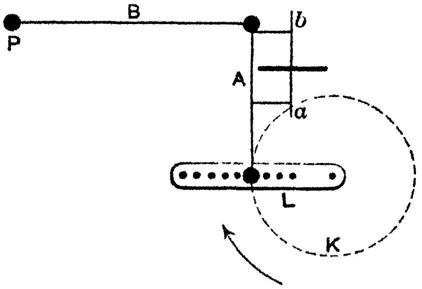

FIG. 19.

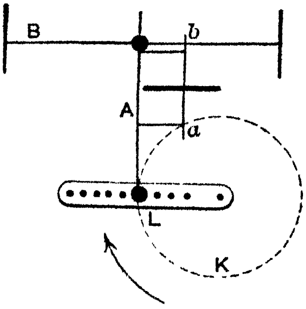

FIG. 20.

The instrument is now placed in the position shown in Figs. 19 and 20 when the circle *K* appears on the *right*-hand side of the tracing-arm A, and the passage of the tracer takes place in exactly the same way.

If the results obtained right and left of the tracing-arm be equal to one another, it is clear that the axis ab of the measuring-wheel is parallel to the tracing-arm, and, this being so, the second test may now be applied. But if the result be *greater* in the first case, that is to say, when the circle lies to the left of the tracing-arm, the extremity a of the axis of the measuring-wheel must be farther removed from the tracing-arm; if it be *less*, that extremity must be brought nearer to the tracing-arm.

Second Test. The tracing-arm is adjusted by means of the vernier on the guide and by means of the micrometer-screw, in accordance with the formulæ for different areas; it then is fixed within the guide by means of the binding-screw. The circumferences of circles of various sizes are then traveled over with the check-rule, and the results thus obtained are multiplied into the unit of the vernier corresponding to the area given for that particular adjustment by the formula. The figures thus obtained ought to be equal to the calculated area of the circles included by the circumferences. If the results obtained with the planimeter fall short of the calculated areas to the extent of $\frac{1}{n}$ of those areas, the length of the tracing-arm, that is to say, the distance between the tracer and the fulcrum of the tracing-arm, must be reduced to the extent of $\frac{1}{n}$ of that length; in the opposite case it must be increased in the same proportion. The vernier on the guide-piece of the tracing-arm shows the length thus defined with sufficient accuracy, usually in half-millimeters, or about fiftieths of an inch, on the gauged portion of the arm.

In order to test the accuracy of the readings according to the two methods just described, some prefer the use of a check-plate in lieu of the check-rule. The check-plate is a circular brass disk upon which are engraved circles with known radii.

It is advisable to apply the second test also to a large diagram drawn on paper and having a known area.

The instrument having been found correct or its errors determined, it may now be used with confidence.

The following form is used to record the results of the test:

DEPARTMENT OF EXPERIMENTAL ENGINEERING.
SIBLEY COLLEGE.

Calibration of .... Planimeter.
Length of Tracing Arm = l = ... Ithaca, N. Y., ... 19...
Diameter of Register Wheel = d = ... By ....
% Error from Calibration ....
% Error from $l\pi d$ = ...

PARALLELISM. (First Test.)

No.	Instrument Reading.	Area from Instrument.	No.	Instrument Reading.	Area from Instrument.
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....

ACCURACY (Second Test.)

No.	Instrument Reading.	Area from Instrument	Actual Area	Length of Arm in Error, %.	
				Too long	Too short.
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....

ZERO-CIRCLE.

No.	Instrument Reading	Area from Instrument.	Guide Circle Dia.	Computed Area	Area of Zero-Circle.
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....

32. Errors of Different Planimeters. — Professor Lorber, of the Royal Mining Academy of Loeben, in Austria, made extensive experiments on various planimeters, with the results shown in the following table:

Area in —		The error in one passage of the tracer amounts on an average to the following fraction of the area measured by —				
		The ordinary Polar Planimeter Unit of Vernier 10 sq mm = .0155 sq in	Stark's Linear Planimeter Unit of Vernier 1 sq mm = .0015 sq in	Suspended Planimeter Unit of Vernier 1 sq mm = .0015 sq in	Rolling Planimeter —	
Square cm	Square inches				Unit of Vernier 1 sq mm = .0015 sq in	Unit of Vernier 1 sq mm = .0001 sq in
10	1.55	$\frac{1}{75}$	$\frac{1}{588}$	$\frac{1}{625}$	$\frac{1}{625}$	$\frac{1}{1000}$
20	3.10	$\frac{1}{148}$	$\frac{1}{1000}$	$\frac{1}{1111}$	$\frac{1}{1000}$	$\frac{1}{2000}$
50	7.75	$\frac{1}{353}$	$\frac{1}{1852}$	$\frac{1}{2500}$	$\frac{1}{2000}$	$\frac{1}{3000}$
100	15.50	$\frac{1}{682}$	$\frac{1}{2857}$	$\frac{1}{4177}$	$\frac{1}{3333}$	$\frac{1}{5000}$
200	31.00	$\frac{1}{1274}$	$\frac{1}{1257}$	$\frac{1}{7143}$	$\frac{1}{5128}$	$\frac{1}{7692}$
300	46.50			$\frac{1}{9375}$	$\frac{1}{8000}$	$\frac{1}{10000}$

The absolute amount of error increases much less than the size of the area to be measured, and with the ordinary polar planimeter is nearly a constant amount.

The following table is deduced from the foregoing, and shows the error per single revolution in square inches:

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Area in —		Error in one passage of the tracer in square inches			
		Polar Planimeter Unit of Vernier 10 sq mm = .0155 sq in	Suspended Planimeter Unit of Vernier 1 sq mm = .0015 sq in	Rolling Planimeter	
Square cm	Square inches			Unit 1 sq mm = .0015 sq in	Unit of Vernier 1 sq mm = .0001 sq in
10	1.55	0.0207	0.0025	0.0025	0.00155
20	3.10	0.0216	0.0028	0.0031	0.00158
50	7.75	0.0221	0.0031	0.0038	0.00258
100	15.50	0.0227	0.0035	0.0043	0.00310
200	31.00	0.0243	0.0043	0.0060	0.00403
300	46.50		0.0049	0.0058	0.00465

These errors were expressed in the form of equations, as follows, by Professor Lorber. Let *f* equal the area corresponding to one complete revolution of the record-wheel; let *dF* be the error in area

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due to use of the planimeter. Then for the different planimeters we have the following equations:

Lineal planimeter,	$dF = 0.00081\,f + 0.00087\,\sqrt{Ff}$;
Polar planimeter,	$dF = 0.00126\,f + 0.00022\,\sqrt{Ff}$;
Precision polar planimeter,	$dF = 0.00069\,f + 0.00018\,\sqrt{Ff}$;
Suspended planimeter,	$dF = 0.0006\,f + 0.00026\,\sqrt{Ff}$;
Rolling planimeter,	$dF = 0.0009\,f + 0.0006\,\sqrt{Ff}$.

33. **Special Planimeters.** — These, much more complicated than those described, have been made for special purposes. Among these we may mention Amsler's mechanical integrator for finding the moment of inertia, and Corradi's mechanical integraph for drawing the derivative of any curve, the principal curve being known, thus giving a graphic representation of moment.

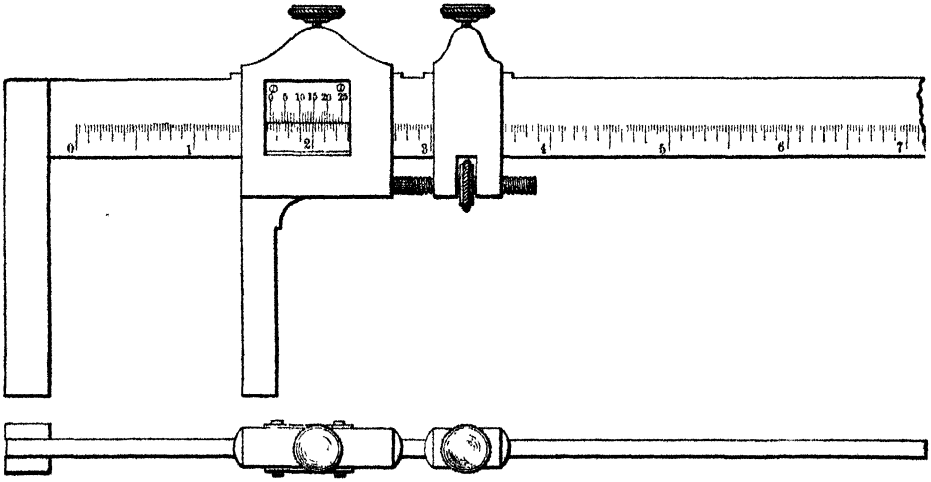

FIG. 21. — VERNIER CALIPER.

34. **Vernier Caliper.** — This instrument consists of a sliding-jaw, which carries a vernier, and may be moved over a fixed scale. The form shown in Fig. 21 gives readings to $\frac{1}{40}$ inch on the limb, and $\frac{1}{25}$ this amount or to one-thousandth of an inch on the vernier. The reading

of the vernier as it is shown in the figure is 1.650 on the limb and 0.002 on the vernier, making the total reading 1.652 inches. This instrument is useful for accurate measurements of great variety; the special form shown in the cut has a heavy base, so that it will stand in a vertical position and may be used as a height-gauge. To use it as a caliper, the specimen to be measured is placed between the sliding-jaw and the base; the reading of the vernier will give the required diameter.

35. **The Micrometer.** — This instrument is used to determine dimensions when great accuracy is desired. It consists of a finely cut screw, one revolution of which will advance the point an amount equal to the pitch of the screw. The screw is provided with a graduated head, so that it can be turned a very small and definite portion of a revolution. Thus a screw with forty threads to the inch will advance for one complete revolution $\frac{1}{40}$ of an inch, or 25 thousandths. If this be provided with a head subdivided to 250 parts, the point would be advanced one ten-thousandth of an inch by the motion sufficient to carry the head past one subdivision.

The micrometer is often used in connection with a microscope having cross-hairs, and in such a case represents the most accurate instrument known for obtaining the value of minute subdivisions; it is also often used in connection with the vernier. The value of the least reading is determined by ascertaining the advance due to one complete revolution and dividing by the number of subdivisions. The total advance of the screw is equal to the advance for one revolution multiplied by the number of revolutions plus the number of subdivisions multiplied by the corresponding advance for each.

The accuracy of the micrometer depends entirely on the accuracy of the screw which is used.

Accuracy of Micrometer-screws.— The accuracy attained in cutting screws is discussed at length by Professor Rogers in Vol. V. of "Transactions of American Society of Mechanical Engineers," from which it is seen that while no screw is perfectly accurate, still great accuracy is attained. The following errors are those in one of the best screws in the United States, expressed in hundred-thousandths of an inch, for each half-inch space, reckoned from one end.

CORNELL UNIVERSITY SCREW.

Total Errors in Hundred-thousandths of an Inch.

No of Space	Total Error	No. of Space	Total Error	No of Space	Total Error.
0	0	12	−4	24	−8
1	+6	13	−7	25	−7
2	+8	14	−9	26	−7
3	+9	15	−7	27	−9
4	+7	16	−10	28	−9
5	+9	17	−11	29	−7
6	+7	18	−11	30	−7
7	+4	19	−10	31	−6
8	+5	20	−10	32	−7
9	0	21	−9	33	−7
10	−1	22	−11	34	−3
11	−2	23	−10	35	−2
				36	0

An investigation made by the author on the errors in the ordinary Brown and Sharpe micrometer-screw failed to detect any errors except those of observation, which were found to be about 4 hundred-thousandths of an inch for a distance equal to three-fourths its length. The errors in the remaining portion of the screw were greater, the total error in the whole screw being 12 hundred-thousandths of an inch. As the least reading was one ten-thousandth, the screw was in error but slightly in excess of the value of its least subdivision. In another screw of the same make the error was three times that of the one described.

36. The Micrometer Caliper consists of a micrometer-screw shown in Fig. 22, which may be rotated through a fixed nut. To the screw is attached an external part or *thimble*, which has a graduated edge subdivided into 25 parts. The fixed nut is prolonged and carries a cylinder, termed the *barrel*, on which are cut concentric circles, corresponding to a scale of equal parts, and a series of parallel lines, which form a vernier with reference to the scale on the thimble, the least reading of which is one-tenth that on the thimble. If the screw be cut 40 threads per inch, one revolution will advance the point 0.025 inch; and if the thimble carry 25 subdivisions, the least reading past any fixed mark on the barrel would be one-thousandth of an inch.

By means of the vernier the advance of the point can be read to ten-thousandths of an inch. Thus in the sketches of the barrel and thimble scales in Fig. 22 the zero of the vernier coincides in the upper

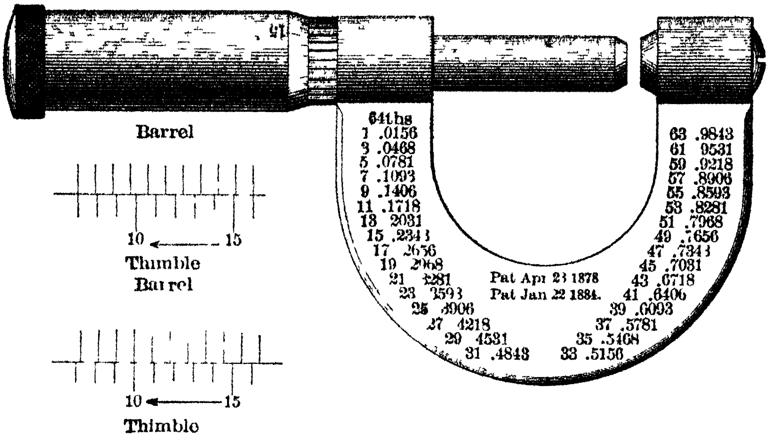

FIG. 22.—MICROMETER CALIPER.

sketch with No. 7 on the thimble; but in the lower figure the zero of the vernier has passed beyond 7, and by looking on the vernier we see that the 4th mark coincides with one on the thimble, so that the total reading is 0.007 + 0.0003, which equals 0.0073 inch.

This number must be added to the scale reading cut on the barrel to show the complete reading. The principal use of the instrument is for measuring external diameters less than the travel of the micrometer screw, usually less than 2 inches in this type.

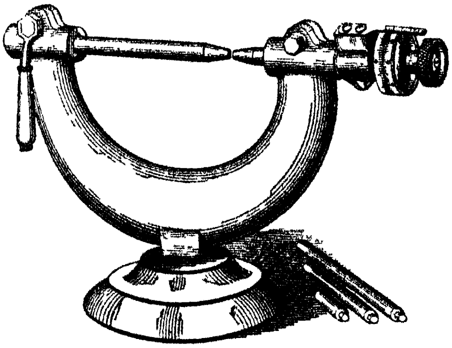

FIG. 23.—SWEET'S MEASURING-MACHINE.

The Sweet Measuring-machine.—The Sweet measuring-machine is a type of micrometer caliper arranged for measuring larger diameters than the one previously described. The general form of the instrument is shown in Fig. 23. The micrometer-screw has a limited range of motion, but the instrument is furnished with an adjustable tail spindle, which is set at each observation for distances in even inches, and the micrometer-screw is used only to measure the fractional or decimal parts of an inch. The instrument is furnished with an external scale, graduated on the upper edge to read in binary fractions of an inch and on the lower edge to read in decimals of an inch; this scale can be set at a slight angle with the axis to correct for any error in the pitch of the micrometer-screw. The graduated disk is doubly graduated, the right-hand graduations corresponding to those on the lower side of the scale. The scale and graduated disk are shown in Fig. 24, and the readings corresponding to the positions shown in the figure are 0.6822, the last number being estimated.

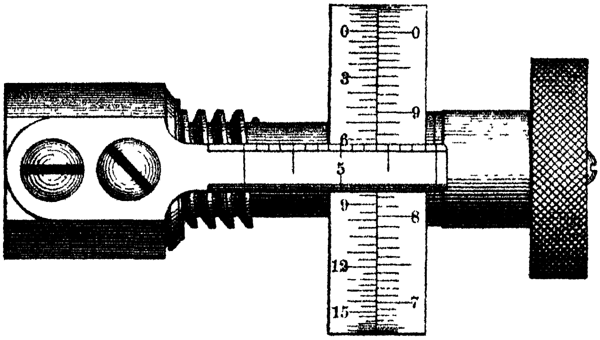

FIG. 24.

The back or upper side of the scale, and the left-hand disk, are for binary fractions, the figures indicating 32ds. Fig. 25 shows the arrangement of the figures. Beginning at 0 and following the line of cords to the right, the numbers are in regular order, every fifth one being counted, and coming back to 0 after five circuits. This is done to eliminate the factor five from the ten-thread screw. In Fig. 24 the portion to the left of 0 in Fig. 25 is seen.

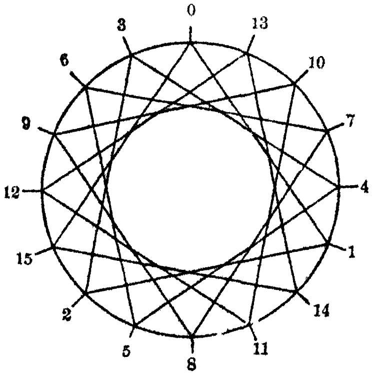

FIG. 25.

The back side of the index-bar is divided only to 16ths, the odd 32ds being easily estimated, as this scale is simply used for a "finder," thus: In the figure the reading line is very near the $\frac{11}{16}$ mark, or *six* 32ds beyond the half-inch. This shows that 6 is the significant figure upon this thread of the screw. The other figures belong to other threads. The figure 6 is brought to view when the reading line comes near this division of the scale. Bring the 6 to the front edge of the index-bar, and the measurement is exactly $\frac{11}{16}$ *without any calculation.* Thus every 32d may be read, and for 64ths and other binary fractions take the nearest 32d below and set by the intermediate divisions, always remembering that it requires *five* spaces to count *one.*

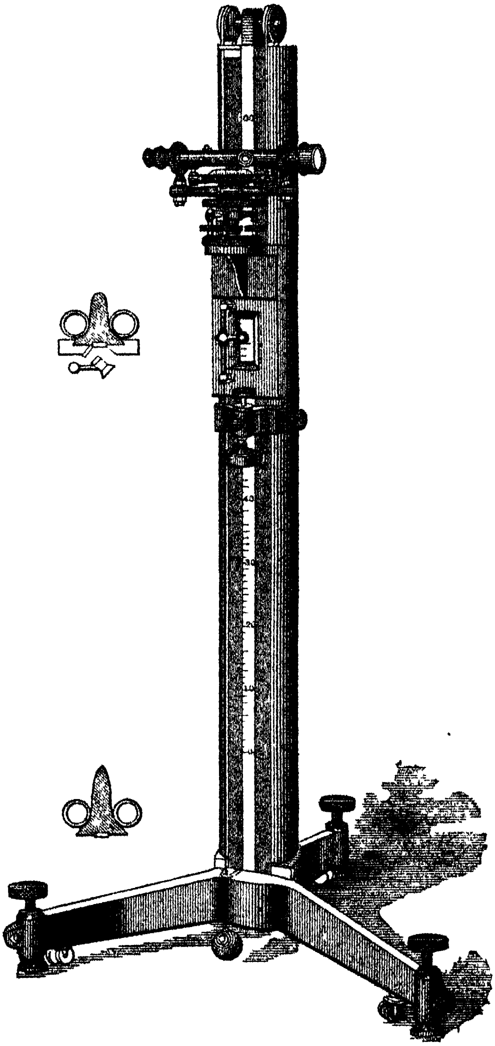

FIG. 26. — CATHETOMETER.

37. The Cathetometer. — This instrument is used extensively to measure differences of levels and changes from a horizontal line. Primarily it consists of one or more telescopes sliding over a vertical scale, with means for clamping the telescope in various positions and of reading minute distances. The one shown in the engraving (Fig. 26) consists of a solid brass tripod or base supporting a standard of the same metal, the cross-section of which is shown at different points by the small figures on the left. A sliding-carriage, upon which is secured the small leveling instrument and which has also a vernier scale as shown, is balanced by heavy lead weights, suspended within the brass tubes on either side by cords attached to the upper end of the carriage, and passing over the pulleys shown at the top of the column. The column is made vertical by reference to the attached plumb-line.

The movable clamping-piece below the carriage is fixed at any point required, by the screw shown at its side, after which the telescope can be raised or lowered by rotating the micrometer-screw attached to the clamp. The telescope is provided with cross-hairs, which can be adjusted by reversing in the wyes and turning 180 degrees in azimuth. The vertical scale is provided with vernier and reading-microscope.

38. **Aids to Computation** may be of three kinds: graphical, tabular, or mechanical.

Graphical methods for multiplying and dividing are usually given in treatises on geometry and are often sufficiently accurate for engineering purposes.

Tabular aids are of two kinds: arithmetic, such as product, quotient and reciprocal tables, and logarithmic. The Rechentafeln of A. L. Crelle give one million products and will be found of much value in multiplication and division. There are a number of good logarithmic tables, the standard being those of Vega, but any good four or five place table is sufficient for engineering work. As compared with graphical methods, the use of tables usually gives more accurate results with a smaller expenditure of time.

Mechanical aids may be either computing machines or slide-rules. Several very excellent machines for multiplying and dividing which give accurate results to from 14 to 17 places are now made. Of these we may mention the calculating machine of George B. Grant of Boston; the Brunsvega of Grimme-Natalis & Co., Brunswick, Germany; the Comptometer, made by the Comptometer Co. of Chicago; and the adding machine made by the Burroughs Adding Machine Co., Detroit, Mich. Slide-rules of compact form but with scales 40 feet in length, as designed by Thatcher or Fuller, can also be obtained of the principal stationers.

The processes of arithmetical calculation are almost entirely mechanical and involve no reasoning powers, yet they are of utmost importance in connection with experimental work. Unless the observations of the experiment are correctly recorded and the necessary calculations for expressing the result made accurately, the experimental work will either be of no value, or, what is worse, positively misleading. For these reasons mechanical methods of computation, which involve at best small errors of known magnitude, are to be adopted whenever possible in reducing engineering experiments.

The calculating machine is of special value, since if the mechanical processes are correctly performed the results will be given with accuracy for the number of places within limits of the machine. Numerous calculating machines have been designed, the most noted of which is the "difference engine" designed by Babbage in 1822 and on which the English Government expended more than $85,000 without bringing it to perfection. The first practical machine which accomplished anything worthy of permanent record was invented by Thomas de Colmar in 1850, and since that time numerous others, designed on similar lines, have appeared, of which should be mentioned those invented by Tate, Burkhardt, Grant, Baldwin, and Odhner. The Grant machine, developed from 1874 to 1896, had reached a high degree of perfection, but is no longer sold. The Odhner or Brunsvega, referred to above, was shown at the World's Fair in 1893, and differs from the Grant principally in the arrangement of parts, and in the fact that, as now sold, it possesses an index or counter to register the multiplier during the process of multiplication.

In the Brunsvega the result is read on a series of wheels arranged on the same axis and so connected that ten revolutions of one of lower denomination are required for one of the next higher, etc., these wheels being readily and simultaneously set at zero. Series of digits from 1 to 9 are engraved in vertical parallel columns on a keyboard. By setting a lever opposite any number and turning a crank once, the number will appear on the result-wheels; by turning the crank twice, the result-wheels will show twice the number, etc.

The shaft carrying the result-wheels can be shifted several places, so that it is possible to multiply by numbers of any denomination with less than ten revolutions of the crank. Subtraction is performed by starting with the larger number on the result-wheels and the smaller number on the keyboard and revolving the crank in the opposite direction from that required for addition. Dividing is done as a sort of continued subtraction, and is a more complicated operation. The machine is readily worked as a difference engine, thus permitting its use for computing complicated tables.

A trial made in the U. S. Coast Survey of the relative rapidity and accuracy of the Grant calculating machine and a seven-place table of logarithms, in multiplying seven figures by seven figures and retaining seven figures in the result, showed the average time of multiplication with the machine as 56 seconds, and with logarithms 157 seconds; the number of errors in 100 trials, with the machine 7, with logarithms 12. A trial made at Sibley College showed more favorable for the machine, probably because the observers were not so expert with logarithms.

CHAPTER III.

STRENGTH OF MATERIALS.— GENERAL FORMULÆ.

In this chapter a statement is made of the principal definitions and formulæ required for the experimental work in "Strength of Materials." The full demonstration of the formulæ and discussion of "Strength of Materials" is to be found in "Mechanics of Engineering," by I. P. Church, or by Mansfield Merriman; "Strength of Materials," by D. V. Wood; "Materials of Construction," by R. H. Thurston; "Materials of Construction," by J. B. Johnson; "The Testing of Materials of Construction," by Unwin; "Materialienkunde für den Maschinenbau," by A. Martens.

The object of experiments in "Strength of Materials" is to ascertain the resistance of different materials to stresses of various character; to find the characteristics distinguishing the different qualities, *i.e.*, the good from the bad; to obtain experimental proof of laws deduced theoretically; and to discover the general laws of variation of properties as dependent on form, kind, and quality of material.

The more general tests may be classified under the heads of: (1) tension; (2) compression; (3) transverse loading; (4) direct shear; (5) torsion; (6) impact; (7) repeated loading and unloading, or fatigue testing.

39. **Definitions.** --*Stress* is the force applied to a material. *Stress*, used in the sense of *stress intensity*, means force per unit area, or in English units, pounds per square inch. Stresses are fundamentally of two kinds: *normal stress*, conceived of as acting perpendicular to an imagined exposed section of the material; and *tangential* or *shearing stress*, acting parallel to an exposed section.

Deformation is the distortion or change of shape of a material which accompanies the action of a stress. *Deformation* is mainly *elongation* or *compression*, if the stress is normal; or is *angular*, if the stress is a shear. (See Figs. 27 and 28.) With normal stress, unit deformation is measured in inches change of length per inch of original length, or in percentage of the original length. With shear stress angular deformation is measured in π (radian) measure.

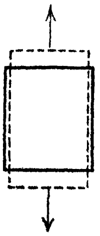

FIG. 27. — NORMAL STRESS ELONGATION.

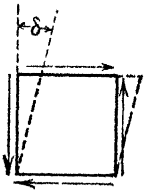

FIG. 28. — SHEAR STRESS ANGULAR DEFORMATION.

Up to a certain point, called the *elastic limit,* stress and deformation are proportional to each other. If stressed within the limit, once or many times, the recovery to original size and shape is perfect when the load is taken off. If the stress exceeds the elastic limit the material experiences a permanent change of shape and no longer returns entirely to its original dimensions when the stress is removed. Such permanent deformation is called a *set.*

Many materials can be stressed far beyond the elastic limit, undergoing during this stressing a considerable permanent deformation. Such materials are called *ductile.* The *ultimate strength* of a material is the maximum load the piece carries before breaking, divided by the *original* area. The *elastic limit strength* is similarly found from the load at the elastic limit. The *ductility,* for tension loading, is the permanent set at the breaking load expressed as a percentage of the original length. The *reduction of area* is the percentage difference between initial and final areas of section of the tension test piece, computed on the initial area.

There are two important secondary quantities depending on both stress and deformation. The *modulus of elasticity* measures the stiffness of a material; it is equal to the ratio of stress to corresponding deformation, in unit quantities, under conditions of elastic loading. The resilience measures the work which the material can absorb in being stressed to a certain value. So long as the loading is elastic, the resilience of a piece is the half product of load and deformation of the piece. The *modulus of resilience* of the material is the work done upon unit volume (one cubic inch) in stressing to the elastic limit; it is the half product of elastic limit strength by corresponding unit deformation.

Working loads must always be within the elastic limit. The ratio of ultimate strength to working stress is called the *factor of safety*. It measures the safety of a structure against rupture. The ratio of elastic limit strength to working stress measures the safety for continued operation of the structure.

40. Stress-deformation Diagrams are curves showing the relations between stress and deformation as load is applied to a piece of material. These diagrams show by inspection the general nature of a material. The diagrams are plotted with deformations as abscissæ and stresses as ordinates. The general form of the diagram for certain typical materials is shown in Figs. 29 to 31.

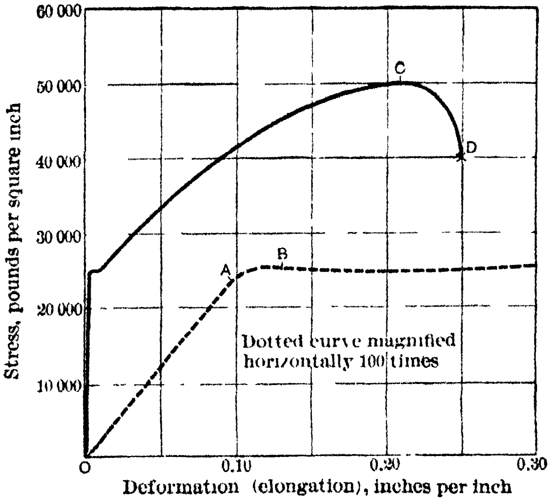

FIG. 29.

A, Fig. 29, marks the *elastic limit* found only in worked materials; *B*, Figs. 29 and 30, is the *yield point*, where the material suddenly begins to take large permanent set; *C* is the ***maximum*** and *D* the ***breaking*** load. The *modulus* of *elasticity* is the slope of the *elastic line OA*, Fig. 29. The *modulus* of *resilience* is represented by the area bounded by the axis of abscissas, the elastic line *OA*, and a perpendicular dropped from *A* to the axis of abscissas. In commercial practice, *B*, the yield point, is taken as the practical elastic limit in the case of ductile materials; for brittle materials, as cast iron, either the breaking point, *C–D* (Fig. 31), or some arbitrarily chosen point on the curved load line, is taken as the working elastic limit.

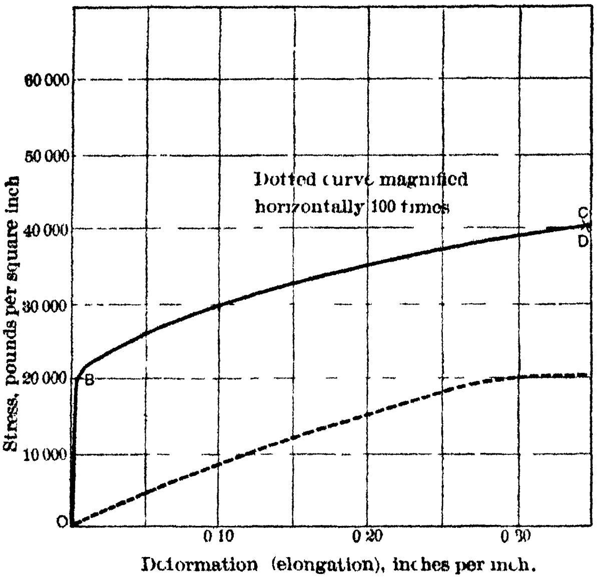

FIG. 30.

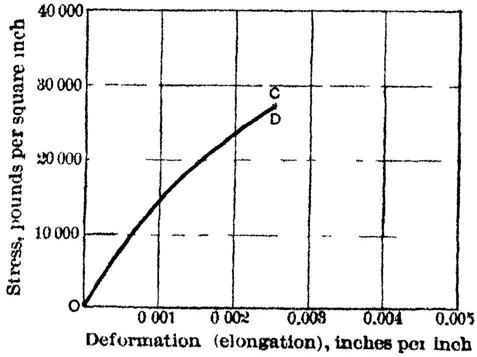

FIG. 31.

41. Notation. — The notation used is similar to that of Church's "Mechanics," and is as follows:

Quantity.	Symbol.		
	General	Elastic limit	Maximum.
Load applied	P	P'	P_m
Stress intensity	p	p'	p_m
Total elongation or compression	λ	λ'	λ_m
Increment of elongation	$\Delta\lambda$	..	..
Relative elongation	ε	ε'	ε_m
Bending or twisting moment	M	M'	M_m
Total angle of torsion	α	α'	α_m
Relative shear distortion	δ	δ'	δ_m
Vertical shear, total	J or S	J'	J_m
Shear intensity	s	s'	s_m

	Tension.	Compression	Shear
Modulus of elasticity	E_t	E_c	E_s
Modulus of resilience	U_t	U_c	U_s

Area of section, square inches	F
Original length, inches	l
Breadth of section, inches	b
Height of section, inches	h
Diameter of a circular section, inches	d
Radius of a circular section, inches	r
Rectangular moment of inertia, about gravity axis of section	I
Polar moment of inertia, about gravity center of section	I_p
Maximum fibre distance	e

42. Formulæ for Tension Loading. — Since in a properly constructed tension test-piece the stress is uniformly distributed, we have

$$p = \frac{P}{F}; \tag{1}$$

$$\varepsilon = \frac{\lambda}{l}; \tag{2}$$

$$E_t = \frac{p}{\varepsilon} = \frac{Pl}{F\lambda}; \tag{3}$$

so long as the loading remains elastic;

$$U_t = \tfrac{1}{2} p'\varepsilon' = \tfrac{1}{2} (p')^2 \div E_t. \tag{4}$$

43. Formulæ for Compression Loading. — For *short columns*, or those pieces which fail by crushing down without bending, the formulæ are the same as for tension:

$$p = \frac{P}{F}; \tag{5}$$

$$\varepsilon = \frac{\lambda}{l}; \tag{6}$$

$$E_c = \frac{p}{\varepsilon} = \frac{Pl}{F\lambda}; \tag{7}$$

$$U_c = \tfrac{1}{2} p'\varepsilon' = \tfrac{1}{2} (p')^2 \div E_c. \tag{8}$$

For *long columns*, which fail because of the occurrence of bending, the formulæ in common use are four:

The Rankine or Gordon:

$$P_m = Fp_m \div \left(1 + m\beta \frac{l^2}{k^2}\right); \tag{9}$$

The Ritter or Merriman:

$$P' = Fp' \div \left(1 + m \frac{p'}{4\pi^2 E} \cdot \frac{l^2}{k^2}\right); \tag{10}$$

The Johnson:

$$P' = Fp'\left(1 - m \frac{p'}{4\pi^2 E} \cdot \frac{l^2}{k^2}\right); \tag{11}$$

The Euler:

$$P_m = \frac{4\pi^2 EI}{ml^2} = \frac{4\pi^2 E}{m \cdot \frac{l^2}{k^2}} F; \tag{12}$$

In these formulæ P', P_m, p', p_m have the meanings given in Art. 41, p' and p_m referring here to the results of short column tests of the material; β is a constant characteristic of the material; k is the radius of gyration of the cross-section of the column $\left(k^2 = \frac{I}{F}\right)$; m is a factor of end condition, having the numerical value 1 for "square-ended" columns, $\frac{16}{9}$ for one "square end" and one "round end," and 4 for a "round-ended" column. "Square ends" are those which are rigidly held in line at the point of support; "round ends" are free to hinge at the supports.

It will be seen that formulæ (9) and (10) are convertible in form by making $\beta = \frac{p'}{4\pi^2 E}$; but (11) cannot possibly be converted into (9) or (10), and, given the same values of p', E, l and k, (11) will give a lower value of P' than (10). The Euler formula, while mathematically and experimentally true for very long columns, is useless for any actual case, for it gives too high values of P_m if $\frac{l}{k}$ is less than 150 to 200. The Merriman formula is valid over the widest range of $\frac{l}{k}$ values, beginning with short columns and becoming tangent to the Euler at infinity.

Suitable design values of p_m and β in the Rankine formula are as follows:

Material	p_m	β
Cast iron	80,000	1 ÷ 6,400
Wrought iron	36,000	1 ÷ 36,000
Mild steel...	50,000	1 ÷ 50,000
Timber (oak or similar)	7,500	1 ÷ 3,000

44. Transverse Loading. — In pure transverse loading all the external forces applied are perpendicular to the principal dimension — the length — of the material used. As a result of this loading the beam becomes curved. Layers of material parallel to the length of the beam are then distorted, those on the concave side of the beam being shortened and those on the convex side lengthened. The amount of this distortion or deformation is proportional to the distance of a layer from the instantaneous center of curvature, and hence varies in a straight-line law with distance from one curved surface of the beam to the other (Fig. 32). Consequent on this deformation are set up internal tension and compression forces in the direction of the length of the beam. Because there is no external force in the direction of the length, the internal tension and compression forces must balance each other. The only possible distribution of elastic stresses which satisfies the two conditions of (1) linear variation of intensity in passage from one curved surface to the other, and (2) zero resultant in the direction of the length, is shown in Fig. 33. Both stress and deformation are zero at o, which can be shown to be the gravity axis of the cross-section. A surface through o and parallel to the two curved surfaces of the beam is called the *neutral surface*, and its intersection with a cross-section is the *neutral axis* of the section. The curve assumed by the neutral surface due to the forces acting is called the *elastic curve*.

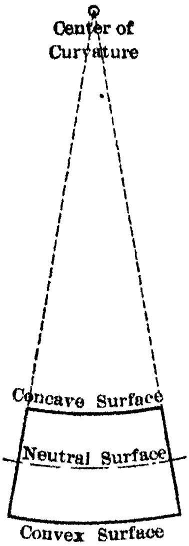

FIG. 32.

The internal stresses (Fig. 33) have a moment about o which balances the moment of external forces about the same point. Writing p for the stress in either outer fiber, e for the distance from neutral axis to outer fiber, I for the rectangular moment of inertia of the cross-section about the neutral axis, the moment $M = \frac{pI}{e}$. The moment M of external forces about any section is found by considering one or the other end of the beam from that section as a free body. M usually varies from point to point of a beam, and as the cross-section dimensions of the beam are usually constant, the only section at which p need be investigated is that corresponding to the maximum value of M.

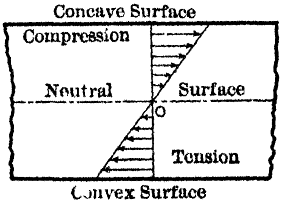

FIG. 33.

Corresponding to the value of p in outer fiber we need a value of ε, the deformation. For this we measure the *deflection* of the beam at that same point of maximum moment at which we compute p. Textbooks on "Mechanics" (see tables below) give values of E in terms of the deflection Δ and load and dimensions of the beam. Using then the relation $\varepsilon = \frac{p}{E}$, the unit deformation ε is readily computed.

By plotting the unit quantities p and ϵ against each other, the curve of the transverse test is the same as that of tension and compression tests, up to the elastic limit. Beyond that point the formulæ are no longer valid. It has been customary, however, to continue to use the elastic formulæ up to the failure of the beam. To distinguish the fictitious values of stress p so computed from a real stress in the material, a special name is given to the value of p at rupture — it is called the *modulus of rupture*. The modulus of rupture is useful in the comparison of one material with another.

In addition to the tension and compression stresses, transverse loading sets up in a beam a system of primary *shears*. These are called the *horizontal and vertical shears*. The *horizontal shear* occurs in surfaces parallel to the neutral surface; the *vertical shear* in planes perpendicular to the neutral surface. At any given point of a section the intensities of horizontal and vertical shear are *identical*. The shears are of zero intensity at either outer surface of the beam and of maximum intensity at the neutral surface. The variation of shear intensity across the section is parabolic if the section does not change in width. The *total vertical shear* at any section is the resultant of external forces perpendicular to the beam on one or the other side of that section. The intensity of primary vertical or horizontal shear stress at the neutral axis is

$$q_m = \frac{J}{Ib_0} \times \left\{ \begin{matrix} \text{area above} \\ \text{(or below)} \\ \text{neutral axis} \end{matrix} \right\} \times \left\{ \begin{matrix} \text{the distance of the center} \\ \text{of gravity of that area} \\ \text{from the neutral axis.} \end{matrix} \right\}$$

J = total vertical shear,

b_0 = width of beam at the neutral axis,

I = rectangular moment of inertia.

For a rectangular cross-section this becomes

$$q_m = \frac{3}{2}\frac{J}{bh},$$

or the intensity is fifty per cent greater than if the shear were uniformly distributed over the cross-section. It is only in the case of

I-beams or similar shapes that the primary shears have sufficient intensity to be a possible cause of failure of the beam.

The following tables will be useful in working up tests:

TABLE I.

Section	Rectangular Moment of Inertia I.	Maximum Fiber Distance e.
Rectangle, width b, depth h	$\frac{1}{12} bh^3$	$\frac{1}{2} h$
Hollow rectangle	$\frac{1}{12} (b_1 h_1^3 - b_2 h_2^3)$	$\frac{1}{2} h_1$
Circle	$\frac{\pi}{4} r^4$	r
Square with side b vertical	$\frac{1}{12} b^4$	$\frac{b}{2}$
Square with side b at 45°	$\frac{1}{12} b^4$	$\frac{b}{2} \sqrt{2}$

TABLE II.

Beam and Loading.	Max Moment and Position. M_m	Max. Deflection and Position. d	Max Total Shear and Position J	Formula for Unit Stress p under Max Moment.	Formula for Unit Deformation ϵ Corresponding to p.
Cantilever-beams fixed at one end.					
1. Loaded at extreme end with P lbs.	Pl at fixed end.	$\frac{Pl^3}{3EI}$ at free end.	P anywhere.	$\left(\frac{le}{I}\right) . P$	$\left(\frac{3e}{l^2}\right) d$
2. Uniform load of intensity $w = W \div l$.	$\frac{Wl}{2}$ at fixed end.	$\frac{Wl^3}{8EI}$ at free end.	W at fixed end.	$\left(\frac{le}{2I}\right) . W$	$\left(\frac{4e}{l^2}\right) . d$
Beams supported at both ends.					
3. Single load P in center.	$\frac{Pl}{4}$ at center.	$\frac{Pl^3}{48EI}$ at center.	Zero at center, elsewhere $P \div 2$.	$\left(\frac{le}{4I}\right) . P$	$\left(\frac{12e}{l^2}\right) . d$
4. Uniform load of intensity $w = W \div l$.	$\frac{Wl}{8}$ at center	$\frac{5Wl^3}{384EI}$ at center.	$\frac{W}{2}$ at ends.	$\left(\frac{le}{8I}\right) . W$	$\left(\frac{48e}{5l^2}\right) . d$

45. Formulæ for Direct Shear. — This stress acts across a piece, without an arm, and tends to produce a square break. It is sensibly uniform over the section, so that

$$S = s \cdot F, \tag{13}$$

where S = total shear, s = intensity of stress, and F = area of section acted on. This stress produces in an element of material an angular distortion which is measured in π measure (see Art. 39 and Fig. 28), and is denoted by δ. The *modulus of shear elasticity* or *modulus of rigidity*

$$= E_s = \frac{s}{\delta}. \tag{14}$$

The *modulus of resilience*

$$U_s = \tfrac{1}{2} s'\delta' = \tfrac{1}{2} \frac{(s')^2}{E_s}. \tag{15}$$

The ultimate shear strength S_m can be obtained by direct experiments, using the specimen in the form of pins or rivets holding links together, the links being held and pulled in a tension-testing machine. A plate can be tested by forcing a punch through, measuring the compression load on the punch. The elastic strength and the angular deformation cannot be measured in the direct testing, but may be accurately determined by tests in torsion, as described below. Direct shear tests must be very carefully conducted, making sure that no arm is given to the load, in order to be satisfactory.

46. Formulæ for Torsion. — The stress produced by torsion is primarily a shear stress on the elements of the material. The torque, or twisting moment (measured in inch-pounds), is applied in planes at right angles to the length of the specimen. The outer fibers of the specimen are twisted into helices, each making an angle with its initial position equal to the angular deformation δ. Any section of the specimen, distant l from the fixed end, is turned in its own plane through an angle α. This angle α is the external deformation which is measured in testing. From the arc which, in Fig. 34, is common to α and δ, we have arc $= e\alpha = l \tan \delta$. Since e and l do not change more than $\frac{1}{10}$ % up to break, $\tan \delta = \frac{e}{l} \cdot \alpha$ may be used throughout the test. If we consider the piece twisted through a certain value of α, the above equation shows that δ varies with c; that is, δ is zero at the center of the section, and increases in direct proportion to distance from the center. Within the elastic limit, stress and deformation are proportional to each other. Hence, until the elastic limit is passed, the shear stress is zero at the center and increases in direct proportion to distance from the center to the maximum in the outer surface of the piece. This variation of stress is similar to that in transverse loading, and leads to a similar formula between moment and stress intensity in outer fiber, viz.:

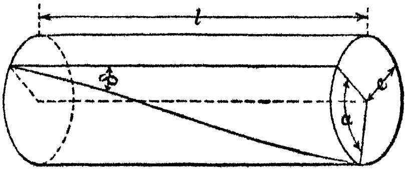

FIG. 34.

$$s = \frac{Mc}{I_p}. \tag{16}$$

For a solid circular section $c = r$ and $I_p = \frac{\pi}{2} r^4$, so that

$$s = \left(\frac{2}{\pi r^3}\right) \cdot M = \left(\frac{16}{\pi d^3}\right) M, \tag{17}$$

where d = diameter of specimen.

$$U_s = \frac{1}{2} s'\delta' = \frac{1}{2} \frac{(s')^2}{E_s}, \tag{18}$$

$$E_s = \frac{s}{\delta} \text{ along an elastic line.} \tag{19}$$

The resilience of the test-piece is $\frac{1}{2} M'\alpha'$. (20)

As in transverse loading, these formulæ are derived for elastic loading and are not valid when the elastic limit is passed. Beyond the elastic limit the values of s computed from $\frac{Mc}{I}$ are higher than the actual values of s in outer fiber of the test-piece. It is customary, however, to continue the computation by the elastic formulæ up to the break. The value of s_m so computed is fictitious, and under the name of *modulus of rupture* in torsion loading is used in comparison of materials.

47. Secondary Stresses. — In the preceding sections the stresses described have been those which are directly produced by the loadings, *i.e.*, what may be called *primary stresses.* Of equal importance to or of greater importance than these primary stresses in determining the yield and failure of materials (but not their elastic action) are the *secondary stresses.*

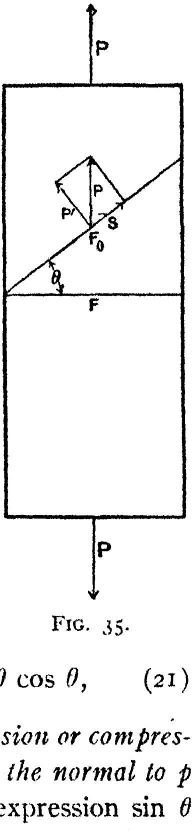

FIG. 35.

In Fig. 35 is represented a body in tension. The intensity of the primary stress is $p = \frac{P}{F}$, P being the total end force and F the area of the normal section. If any other section than the normal be investigated, as F_0 at an angle θ with the normal, it will be found to carry a tension P' and a shear S, so related that the resultant of P' and S is P. P' is less than P (equals $P \cos \theta$) and F_0 is greater than F (equals $F \div \cos \theta$), so that the tension stress perpendicular to F_0 is much less than the stress $p = \frac{P}{F}$ on the normal plane. But the shear stress,

$$s = \frac{S}{F_0} = \frac{P \sin \theta}{F \div \cos \theta} = \frac{P}{F} \sin \theta \cos \theta = p \sin \theta \cos \theta, \qquad (21)$$

may be a cause of failure. *In general, with every tension or compression stress of intensity p there goes at θ degrees with the normal to p a shear stress of intensity* $s = p \sin \theta \cos \theta$. The expression $\sin \theta \cos \theta$ has a maximum value of $\frac{1}{2}$ when $\theta = 45°$. At 45° with p, $s = \frac{p}{2}$.

For instance, in torsion loading of a soft steel the elastic failure in shear is found at about $s = 15{,}000$ pounds per square inch. If the same material be loaded in tension, the secondary shear will reach the critical intensity when the primary tension stress is 30,000 pounds per square inch. The material will break down sharply and begin to elongate; it will show slip lines of failure at 45 degrees to the axis. It is said the steel has passed its yield-point. The *permanent sets* in ductile materials in tension loading are due to failure occurring through the secondary stressing in shear.

In compression loading of brittle materials the problem is complicated by a sliding over each other of surfaces under pressure. Friction then occurs. It can be shown that friction will change the angle of failure from 45 to $45 + \frac{\phi}{2}$ degrees where ϕ is the friction angle for the material sliding on itself. For instance, cast iron on cast iron has a coefficient of friction corresponding to a friction angle of 20 degrees; the angle in cast iron "short columns" between the planes of failure and the normal plane is

$$45^\circ + \frac{20^\circ}{2} = 55^\circ.$$

If the primary stress is a shear, the secondary stresses are tension and compression. The mathematical relation between the primary and the secondary stresses is *not* the same as in the case above, with the direct stress as the primary stress. Consider first a unit cube of material (Fig. 36) to which a shear S_1 is applied. To balance this and prevent translation horizontally there must be on the opposite face an equal and opposite stress S_2. But S_1 and S_2 constitute a couple; to balance the couple requires S_3 and S_4. This shows that when shear is applied as a primary stress the shears come in pairs at right angles to each other. One cannot have a horizontal shear without having at any given point a vertical shear of the same intensity as the horizontal.

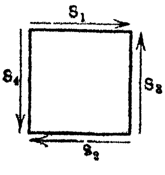

FIG. 36.

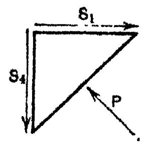

FIG. 37.

Now consider Fig. 37 made from Fig. 36 by cutting on a diagonal. S_1 and S_4 can be balanced by P; the value of P, by summation of vertical components, is

$$P \sin 45^\circ = S_4 = S_1, \text{ or } P = \frac{S_4}{\sin 45^\circ} = \frac{S_1}{\sin 45^\circ}. \qquad (22)$$

If the area on which S_1 or S_4 is applied is F, the area over which P is distributed is $F\sqrt{2}$. The intensity of primary shear is $s = \frac{S_1}{F} = \frac{S_4}{F}$; the intensity of the secondary stress (compression in this case) is

$$p = \frac{P}{F\sqrt{2}} = \frac{S}{F \sin 45 \sqrt{2}} = \frac{S}{F} = s. \qquad (23)$$

If the other diagonal of the cube had been used, we should have found tension for the secondary stress. *When the primary stress is shear of intensity s, there goes with it at* 45 *degrees a secondary stress (tension or compression) of intensity* $p = s$.

An application comes in the case of brittle materials in torsion loading. The tensile strength of cast iron is about 20,000 pounds per square inch. If loaded in torsion, the iron will snap with a brittle fracture at $s = 20{,}000$ pounds per square inch, and the direction of the break will be found to be at 45 degrees with the helices or planes perpendicular to the helices in which shear failure is found. The break has been due to the secondary stressing in tension. The real shear strength of a brittle material cannot be found from torsion loading.

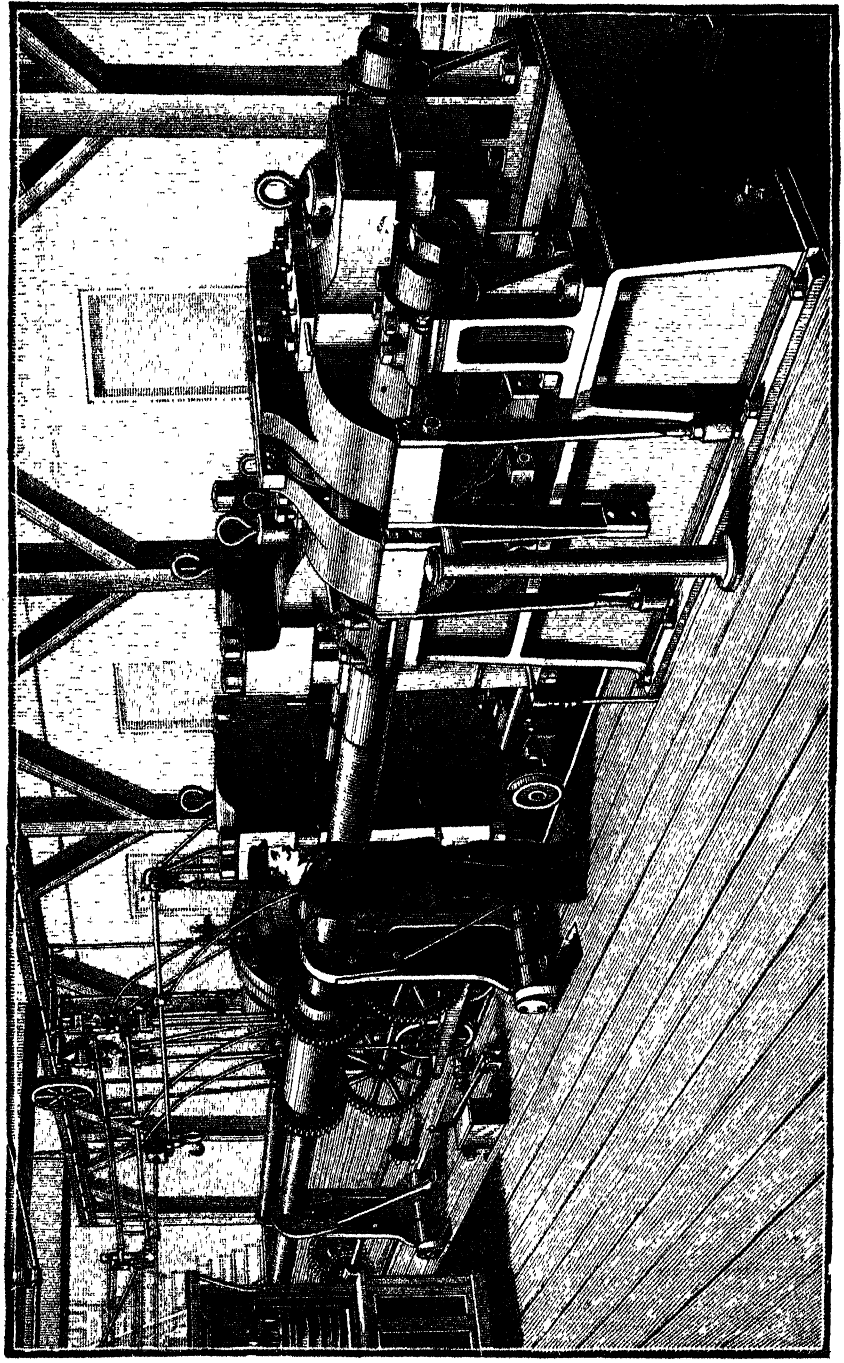

FIG. 38. — THE EMERY TESTING-MACHINE AT THE UNITED STATES ARSENAL AT WATERTOWN.

CHAPTER IV.

STRENGTH OF MATERIALS—TESTING-MACHINES.

48. Testing-machines and Methods of Testing.—The testing-machines consist essentially of, first, a device for weighing or registering the power applied to rupture material; second, head and clamps for holding the specimen; third, suitable machinery for applying the power to strain the specimen; and fourth, a frame to hold the various parts together, which must be of sufficient strength to resist the stress caused by rupture of the specimen. Machines are built for applying tensile, compressive, transverse, and torsional stresses; they vary greatly in character and form; some are adapted for applying more than one kind of stress, while others are limited to a single specific purpose.

In all machines the weighing device should be accurate and sufficiently sensitive to detect any essential variation in the stress, and every laboratory should be provided with means for calibrating testing-machines from time to time; the weighing system is usually independent of the system for applying power, although in certain early machines a single lever mounted on a fulcrum was used, as shown in Figs. 39 and 40, and in which the power system and weighing system were combined, the power applied being measured by multiplying the weight by the ratio of the lever-arms b/a.

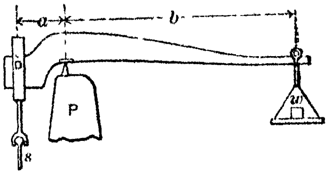

FIG. 39.—OLD FORM.

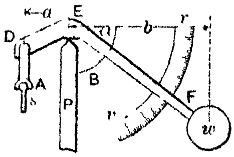

FIG. 40.—THURSTON, POLMEYER.

The power system, when independent of the weighing system, usually consists of a hydraulic press (Fig. 41), or screws driven by a train of gears (Fig. 42). The principal advantage of having the power system independent from the weighing system is due to the fact that under such conditions the stretching of the specimen, which almost invariably takes place, does not affect the accuracy of weighing.

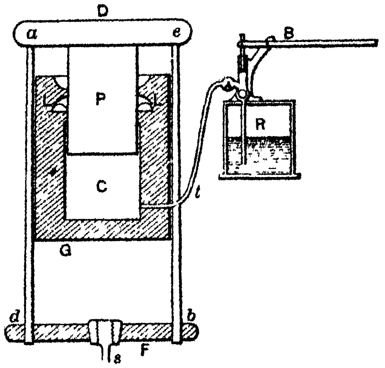

FIG. 41. — HYDRAULIC PRESS.

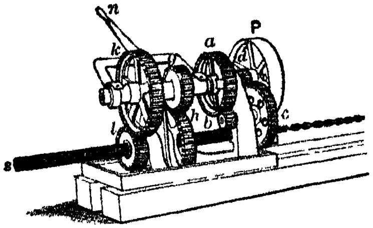

FIG. 42. — FORM OF GEARING.

The shackles or clamps for holding the specimen vary with the stress to be applied. The clamps for tension-tests usually consist of truncated wedges which are inserted in rectangular openings in the heads of the testing-machines, and between which the specimen is placed. The interior face of the wedges is, for flat specimens, plane or slightly convex and serrated, but for round or square specimens is provided with a triangle or V-shaped groove into which the head of the specimen is placed. When the strain is applied to the specimen the wedges are drawn close together, exerting a pressure on the specimen somewhat in proportion to the strain and often injurious to its strength. In many instances shackles with internal cut threads are used, into which specimens provided with a corresponding external thread are screwed; this latter construction is much preferable to the former, though adding much to the expense of preparing the specimen. It is very important that the shackles should hold the specimens firmly and accurately in the axis of the machine and should not exert a crushing strain, which is injurious to the material.

49. General Character of Testing-machines. — Testing-machines are classified as *vertical* or *horizontal*, depending upon the position of the specimen; this, however, is not an important structural difference, although certain classes of machines are better adapted for the one method of testing than the other. Machines may also be classified as *tension*, *compression*, or *transverse* machines, depending upon whether they are better suited to apply one class of stresses than the other, but as the method of testing is generally dependent simply upon the method of supporting the specimen, this classification is of little importance structurally. Machines can perhaps be best classified by the form and character of weighing mechanism, it being generally understood that power may be applied through the medium of gears or by a hydraulic press, as desired, and with any class of machine.

Under this classification we have:

First, the *simple-lever machines*, forms of which have been shown in Figs. 39 and 40, in which the power for breaking was obtained from the weighing mechanism. Fig. 43 shows a single-lever machine much used at the present time in England, in which the power is applied to the specimen at *B*, and the amount of stress is determined by the position of the jockey weight *w* and the amount of weight on the poise *R*.

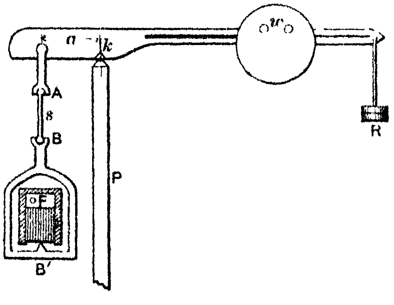

Fig. 43. — Wickstead, Martens, Michaelis, Buckton

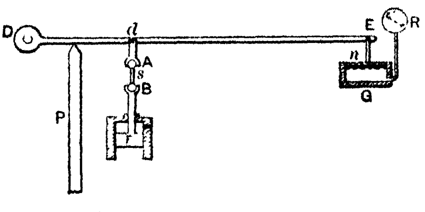

Fig. 44. — Thomasset.

A single-lever machine in which the lever is of the second order is shown in Fig. 44. The specimen is placed between the fulcrum and the weighing mechanism. The latter consists of a hydraulic cylinder with diaphragm and attached gauge, and is interesting as being the prototype of the Emery testing-machine.

Second, *differential-lever machines*, one kind of which is shown in Fig. 45. This consists of a single lever with poise, to which the draw-head is connected by links placed at unequal distances from the fulcrum. A machine of this form was manufactured at one time by Riehlé Brothers.*

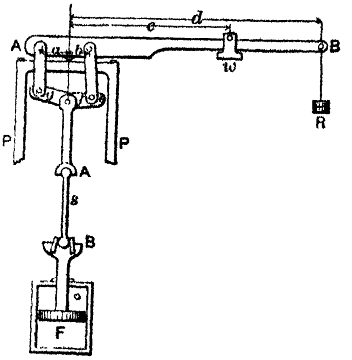

Fig. 45 — Riehlé Hydraulic.

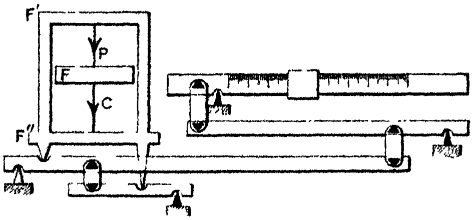

Fig. 46. — Fairbanks Machine.

Third, *compound-lever machines*. These have been much used in America for the last twenty years, and are manufactured by Riehlé Brothers, Olsen, and Fairbanks. In these machines power is usually applied by gearing; at least, such a construction is generally preferred in this country. The diagram, Fig. 46, shows the arrangement of levers adopted in the Fairbanks machine. The movable plate F of the machine, operated by the power device, either exerts a pull P or a compression C upon the stationary plate F', F'', through the medium of the specimen, which in the first case is under tension, in the second case under compression.

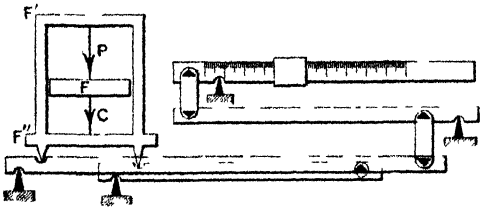

Fig. 47. — Olsen and Riehlé.

Fig. 47 shows arrangement of levers adopted in the Olsen and Riehlé machines, the action being otherwise the same as in the Fairbanks machine.

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* The forces acting in this machine can be represented by the following equation:

$$Rd + wc = \frac{F}{f+g}(af-bg).$$

Fourth, *direct-acting hydraulic machines*. Fig. 48 shows a simple form of a hydraulic machine, in which power is applied by liquid pressure to move the piston *R*, the specimen being located at *s* for tension and at *a′b′* for compression. Machines of this kind have been built of the very largest capacity; for instance, that designed by Kellogg at Athens, Pa., has a capacity of 1,250,000 pounds, and that at the Phœnix Iron Works has a capacity of 2,000,000 pounds, while one built by Professor Johnson at St. Louis has a capacity of about 750,000 pounds. In all these machines the stress is measured by multiplying the readings of the gauge by a constant depending upon the area of the cylinder, the effect of friction being eliminated by keeping the piston rotating, or in other cases neglecting it or determining its amount, and correcting the results accordingly. Such machines are not adapted for accurate testing, but are suited for testing of a character which permits considerable variation from the correct results.

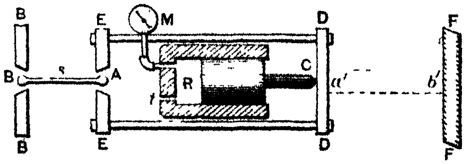

FIG 48 — KELLOGG, JOHNSON.

A modified form of the simple hydraulic machine was designed by Werder in 1852, having a capacity of 100 tons, the principle of its construction being shown in Fig. 49. In this machine the line of action of the stress is in *RF*, while that of the resistance is in the line *Ad* which is to one side of *RF*. These forces are balanced by adjust-

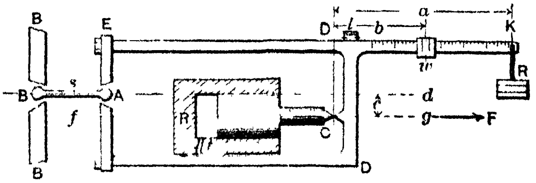

FIG. 49. — WERDER, 1852.

ing the weights on the scale-beam, thus providing means of weighing the force applied to the specimen.

Fig. 50 is a sketch of the working parts of the Maillard machine, in which the weighing apparatus consists of a fluid which is put under pressure by means of a diaphragm against which the stress applied to the specimen reacts. This force is measured on a hydraulic gauge similar in many respects to the weighing apparatus of the Emery testing-machine.

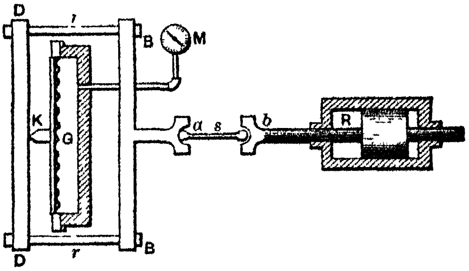

FIG. 50. — MAILLARD.

Fifth, *the Emery machine.* The general principle of the Emery testing-machine is shown in Fig. 51. Power is applied by means of the double-acting hydraulic press R so as to break the specimen either in tension or compression, as desired. The specimen is placed at s, and the stress transmitted is received, if in tension, first by the draw-head BB, thence transmitted to the draw-head $B'B'$, thence in turn to the fluid in the hydraulic support v through a frictionless diaphragm, from which the fluid pressure is transmitted to the vessel with the smaller diaphragm d, the pressure of which is balanced and weighed on the weighing-scale w. If the specimen is in compression

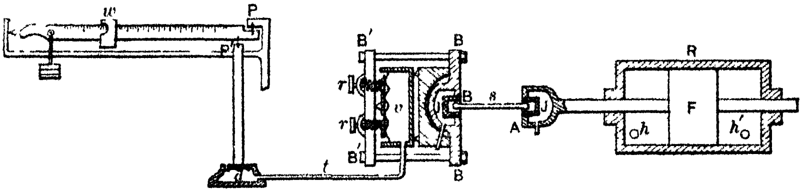

FIG. 51. — EMERY.

the force is transmitted by the draw-head *BB* to the bottom of the hydraulic support *v*, thus crowding the hydraulic support and its contents against the diaphragm, which in turn causes a liquid pressure which is measured on the weighing-scale as before. The springs which receive the pressure of the liquid are adjusted by screws *rr*, connected to the frame, and of sufficient strength to resist the greatest stress applied in compression.

In order that the levers of a testing-machine may transmit the force to the weighing poise with as little loss as possible, and in such a manner that a large force can be balanced by a small weight, a *knife-edge* bearing is in nearly every case provided for each lever. The knife-edge as usually constructed is a piece of hardened steel with a sharp edge which is inserted rigidly in the weighing-lever and rests upon a hardened steel plate fastened to the fulcrum, although in some cases the positions of knife-edge and plate are reversed. The knife-edge should be as sharp as it can be made without crumbling or cutting the contact-plate, and it should be kept clean and free from dirt or rust in order to keep the friction at the lowest possible point. In practice the angle of the knife-edge is made from 30 to 110 degrees, depending upon the load. Machines of the type shown in Fig. 47 have been constructed in which the friction and other losses, as shown by trial, did not exceed 100 pounds in 100,000.

The fulcrums for supporting the levers in the Emery testing-machine are thin plates of steel rigidly connected to both the lever and its support, as shown in Fig. 51. A flexure of the fulcrum-plates is produced by an angular motion of the levers; but as this motion in practice is small, and as the fulcrums are very thin, the loss of force is inappreciable and all friction is eliminated. The plate fulcrums also possess the advantage of holding the levers so that end motion is impossible, and thus preventing any error in weighing due to change of lever-arm. The peculiar form of the plate fulcrums is such as to be unaffected by dirt; furthermore in practice a higher degree of accuracy in weighing has been obtained than is possible with knife-edge levers. The principal characteristics of the Emery machine are, first, the hydraulic supports, which are vessels filled with a liquid and having a flexible side, or diaphragm, which transmits the pressure to a similar support in contact with the weighing apparatus. The detailed construction of a hydraulic support as used in a vertical machine is shown in Fig. 59, its method of operation in Fig. 51. Second, the peculiar steel-plate fulcrums, which have been described. These, together with excellent workmanship throughout, have served to make the Emery testing-machine an instrument of precision with a greater range of capacity and an accuracy far superior to that of any other machine.

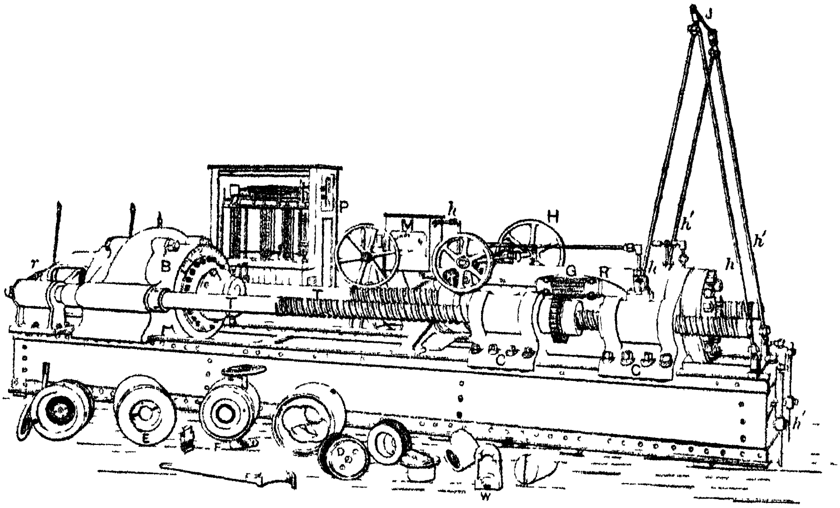

FIG. 52 — EMERY HORIZONTAL MACHINE.

Fig. 52 gives a perspective view of the Emery machine with the working parts marked the same as in the diagram. In this figure *M* is the pump for operating the hydraulic press, *hh'* the connecting piping, *TT* screws forming a part of the frame and used for adjusting the position of the press for different lengths of specimens and of sufficient strength to withstand the shock due to breaking; *P* is the weighing-case, which contains a very elaborate system of weights which can be applied without handling, as described in detail later.

50. Weighing System. — The *weighing system* in the present English machines, and in former ones built in this country, consists of a single lever or scale-beam, along which can be moved a poise, and which can be connected by one or more levers to the test specimen. Such machines are objectionable principally on account of the space occupied.

The weighing device in nearly all recent machines consists of a series of levers, arranged very much as in platform-scales, finally ending in a graduated scale-beam over which a poise is made to move. The machines are usually so constructed that the effect of the strain on the specimen is transmitted into a downward force acting on the platform, and the effect of a given stress is just the same as a given load on the platform.

The weighing-levers usually consist of cast-iron beams carrying hardened steel knife-edges, which in turn rest on hardened-steel bearing plates. This is the system adopted by most scale-makers for their best scales.

In the Emery testing-machines, which are especially noted for their accuracy and sensitiveness, the knife-edges and bearing plates are replaced by thin plates of steel, the flexibility of which permits the necessary motion of the levers.

The weighing device should be *accurate*, and sufficiently sensitive to detect any essential variation in the stress. The amount of sensitiveness required must depend largely on the purposes of the test. An amount less than one tenth of one per cent will rarely make any appreciable difference in the result, and probably may be taken as the minimum sensitiveness needed for ordinary testing. Means should be provided for *calibrating the weighing device*. This can be done, in the class of machines under consideration, by loading the lower platform with standard weights and noting the corresponding readings of the scale-beams. *Testing-machines may be calibrated* with a limited number of standard weights by the use of a test-specimen, which is not to be strained beyond the elastic limit. The weights are successively added and removed, and strain is maintained on the test-piece, equal to the reading on the calibrated portion of the scale-beam.

51. **The Frame.**— The frame of the machine must be sufficiently heavy and strong to withstand the shock produced by a weight equal to the capacity of the machine suddenly applied.

The *weighing levers must sustain* all the stress or force acting on the specimen, without sufficient deflection to affect accuracy of the weighing, and the frame must be able to sustain the shock consequent upon the sudden removal of the load, due to breaking, without permanent set or deflection.

52. Power System. — The power to strain or rupture the specimen is usually applied through the medium of a train of gears or by a hydraulic press, operated by power or hand. The hydraulic machine is very convenient when the stress is less than 50,000 pounds; but if there is any leakage in the valves, the stress will be partially relieved the instant the pump ceases to operate, and difficulty may be experienced in ascertaining the stretch for a given load.

53. Shackles. — The design of shackles or clamps for holding the specimen varies with the strain to be applied. Clamps for tension-tests usually consist of truncated wedges which are inserted in rectangular openings in the heads of the testing-machines and between which the specimen is placed. The interior face of the wedges is for flat specimens plane and serrated, but for round or square specimens it is provided with a triangular or V-shaped groove, into which the head of the specimen is placed. When the strain is applied to the specimen these wedges are drawn closer together, exerting a pressure on the specimen somewhat in proportion to the strain and often injurious to its strength. In tensile testing it is essential to the correct determination of the strength of the specimen that the force shall be applied axially to the material; in other words, it shall have no oblique or transverse component. This requires that the wedge-clamps shall be parallel to the specimen, and that the heads which are as in the clamp shall separate in a right line and parallel to the specimen.

This construction is well shown in the following description of the clamps used in the Olsen and Riehlé testing-machines.

A plan and section of the draw-heads used with the Olsen machine are shown in Fig. 53. *AA* is a counterbalanced lever used to prevent the wedges falling out when the strain is relieved; *BB*, are screws connected to plungers for adjusting the space into which the wedge-clamps are drawn. A lateral motion of the specimen is obtained by unscrewing on one side and screwing up simultaneously on the other side: this adjustment is of advantage in some instances in centering the specimen.

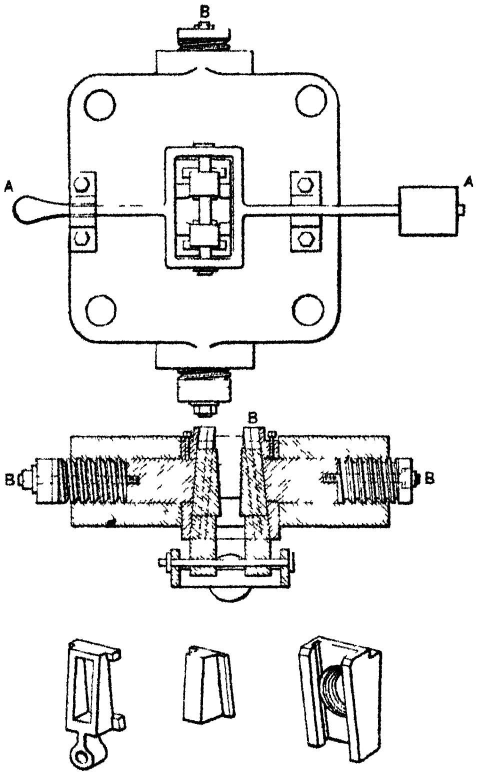

FIG. 53 — DRAW-HEAD OLSEN MACHINE.

The clamps used by Riehlé Brothers for holding flat specimens are shown in Figs. 54 to Fig. 56, as follows:

Fig. 55 is a plan view of the draw-head, with specimen in position: *CC*, curve-faced wedges; *D*, specimen; *A*, draw-head; and *BB*, tension-rods.

Fig. 56 is a sectional view of same. Fig. 54 is a separate view of the wedge. The inclinations of the outside surfaces of the wedges are exaggerated in the drawings, so as distinctly to show the construction.

Wedges have been made with spherical backs, and a portion of the draw-heads mounted on spherical surfaces in order to insure axial strain. Special holders into which screw-threads have been cut have been used with success, and in many instances the specimens have been fastened to the draw-heads by right and left threaded screws.

The wedge-fastening is the most widely used in American commercial testing. It has, however, been found impossible to get a really accurate centering and uniform stressing with this fastening, and in any work of importance screw-ends must be used on the test-piece, with ball and socket supports.

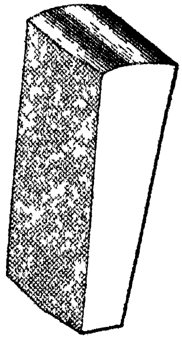

Fig. 54

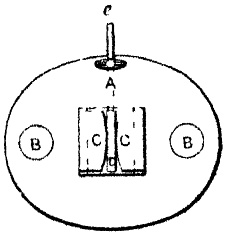

Fig. 55

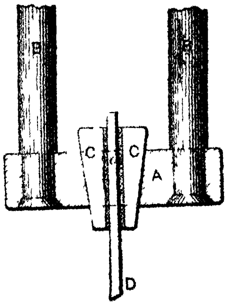

Fig. 56

54. Specifications for Government Testing-machine. — The large machine in use by the United States Government at the Watertown Arsenal was built by Albert H. Emery. The machine is not only of large capacity, but is extremely delicate and very accurate. A perspective view of the machine is shown in Fig. 38.

The requirements of the United States Government as expressed in the specifications, which were all successfully met, were as follows:

1st. A machine with a capacity in tension or compression of 800,000 pounds, with a delicacy sufficient accurately to register the stress required to break a single horse-hair.

2d. The machine should have the capacity of seizing and giving the necessary strains, from the minutest to the greatest, without a large number of special appliances, and without special adjustments for the different sizes.

3d. The machine should be able to give the stresses and receive the shocks of recoil produced by rupture of the specimen without injury. The recoil from the breaking of a specimen which strains the machine to full capacity may amount to 800,000 pounds, instantly

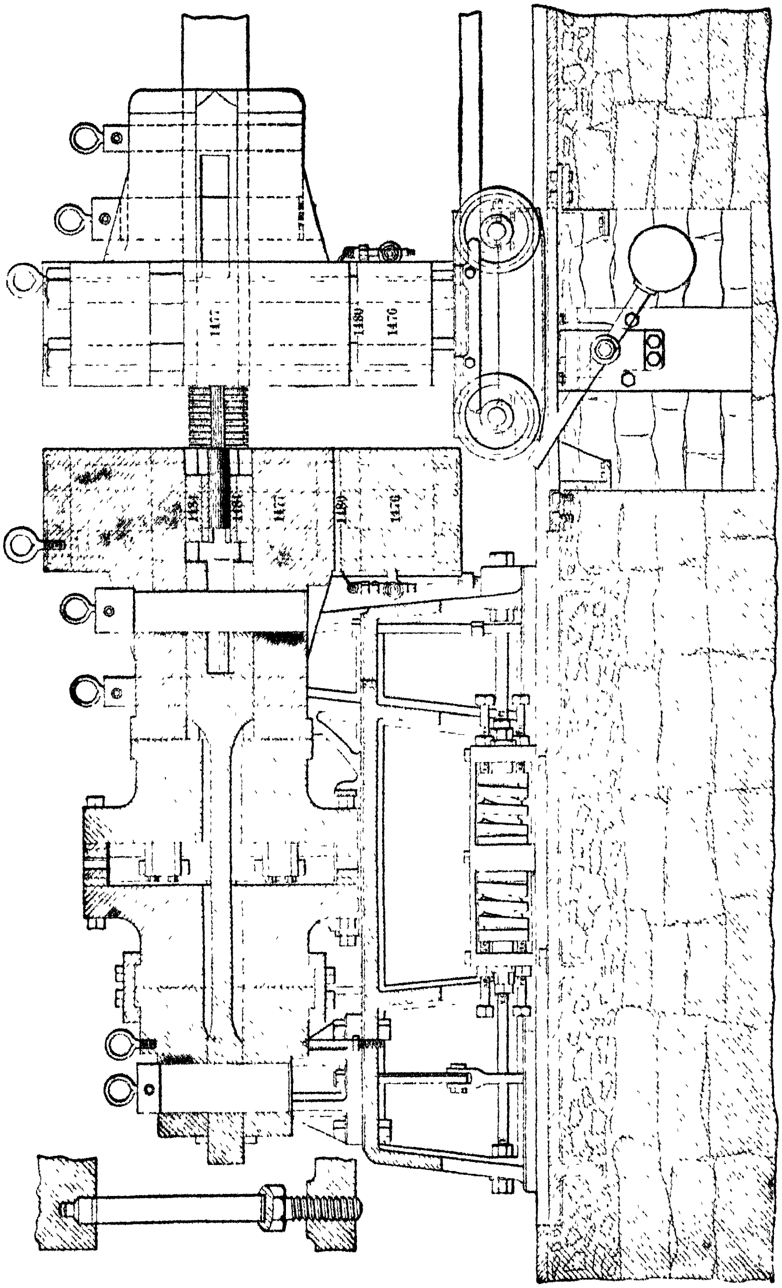

FIG. 57. — LONGITUDINAL SECTION OF WEIGHING HEAD, EMERY TESTING MACHINE.

applied. The machine must bear this load in such a manner as to be sensitive to a load of a single pound placed upon it, without readjustment, the next moment.

4th. The parts of the machine to be at all times accessible.

5th. The machine to be operated without excessive cost.

Description of Emery Testing-machine.—These machines are now constructed by Wm. Sellers & Co. of Philadelphia, under a license from the Yale & Towne Mfg. Co. of Stamford, Conn.

The following description will serve to explain the principle on which the machine acts:

The machine consists of the usual parts: 1. Apparatus to apply the power. 2. Clamps for holding the specimen. 3. The weighing device or scale.

1. The apparatus for applying power consists of a large hydraulic press, which is mounted on wheels as shown in the engravings, Fig. 38 and Fig. 57, and can be moved a greater or less distance from the fixed head of the machine. Two large screws serve to fix or hold this hydraulic press in any position desired, according to the length of the specimen: and when rupture is produced the shock is received at each end of these screws, which tend to alternately elongate and compress, and take all the strain from the foundation.

2. Clamps for holding the specimen. These are peculiar to the Emery machine, and are shown in Fig. 57 in section. This figure also shows a section of the fixed head of the machine, and a portion of the straining-press, with elevation of the holder for the other end of the specimen.

The clamps, numbered 1484 in Fig. 57, are inserted between two movable jaws (1477), which are pressed together by a hydraulic press (1480), resting on the fixed support (1476). By this heavy lateral pressure force equal to 1,000,000 pounds can be applied to hold the specimen. The amount of this force is shown by gauges connected to the press cylinder, and can be regulated as required.

For the vertical machines these shackles or holders are arranged so as to have sufficient lateral motion to keep in the line of the test-piece.

3. The weighing device. This is the especial peculiarity of the Emery machine: instead of knife-edges, thin plates of steel are used

which are flexed sufficiently to allow the necessary motion of the levers. The steel used varies from 0.004 to 0.05 inch thick, and the blades are so wide that the stress does not exceed 40,000 to 60,000 pounds per square inch.

Fig. 58 shows the form of fulcrums used for light forces when the steel fulcrums are in tension.

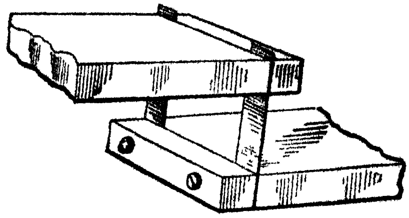

FIG. 58.—EMERY KNIFE EDGE FULCRUM.

The method of measuring the load is practically that of the hydraulic press reversed, but instead of pistons, diaphragms having very little motion are used. Below the diaphragm is a very shallow chamber connected by a tube to a second chamber covered with a similar diaphragm, but of a different diameter. Any downward pressure on the first diaphragm is transmitted to the second, giving a motion inversely as the squares of the diameters. This latter motion may be farther increased in the same manner, with a corresponding reduction in pressure, or it may at once be received by the system of weighing levers. The total range of motion given the first diaphragm in the 50-ton testing-machine is $\frac{1}{400000}$ part of an inch, but the indicating arm of the scales has a motion of $\frac{1}{100}$ of an inch for each pound. This increase of motion and corresponding reduction of pressure is accomplished practically without friction.

The above mentioned parts may be understood from a study of Figs. 59, 60, and 61. Fig. 59 shows the base frame and abutments of the vertical machine. The diaphragm is placed between the frames *EE*, the whole being supported on springs *d*, so as to have an initial tension on the test piece.

The pressure on the diaphragm between the frames *EE* is communicated by the tube *f* to a similar diaphragm in communication with the weighing-levers, Figs. 60 and 61. In case a diaphragm is used it is placed beneath the column *A*, Fig. 61; the motion of the column *A* is communicated to the scale-beams by a system of levers as shown.

The scale-beam mechanism of the testing-machine is so arranged that by operating the handles on the outside of the case the weights required to balance the load can be added or removed at pleasure. The device for adding the weights is shown in Fig. 62. *a*, *b*, *c*, *d*, *e*, and *f* are the weights, which are usually gold-plated to prevent rusting. These when not in use are carried on the supports *A* and *B* by means of pins. When needed, these supports can be lowered by the outside levers, and as many weights as are needed are added to the weighing-poise *CD*.

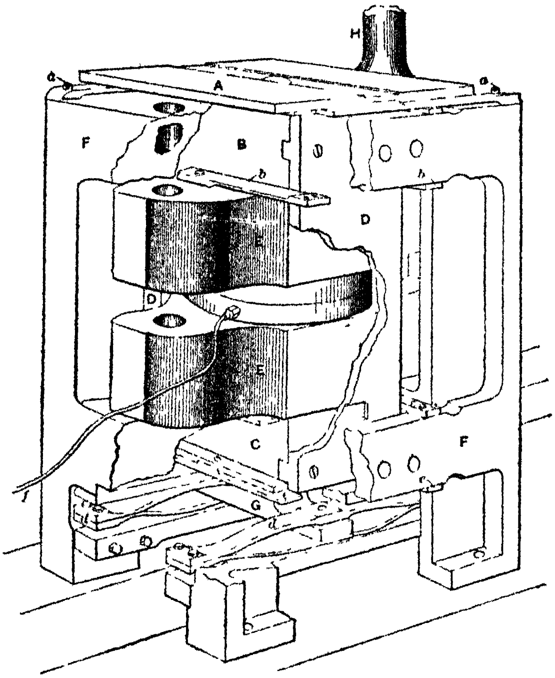

FIG. 59.—THE BASE FRAME AND ABUTMENTS.

55. Riehlé Brothers' Hydraulic Testing-machines.—The testing-machines built by Riehlé Brothers of Philadelphia vary greatly in principles and methods of construction. In the machines built by this firm, power is applied either by hydraulic pressure or by gearing, and the weighing device consists of one or more levers working on steel knife-edges, as in the usual scale construction.

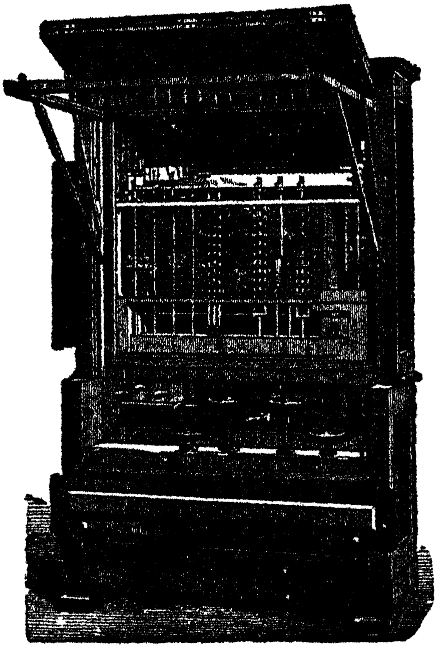

FIG. 60.—SCALE-BEAM AND CASE.

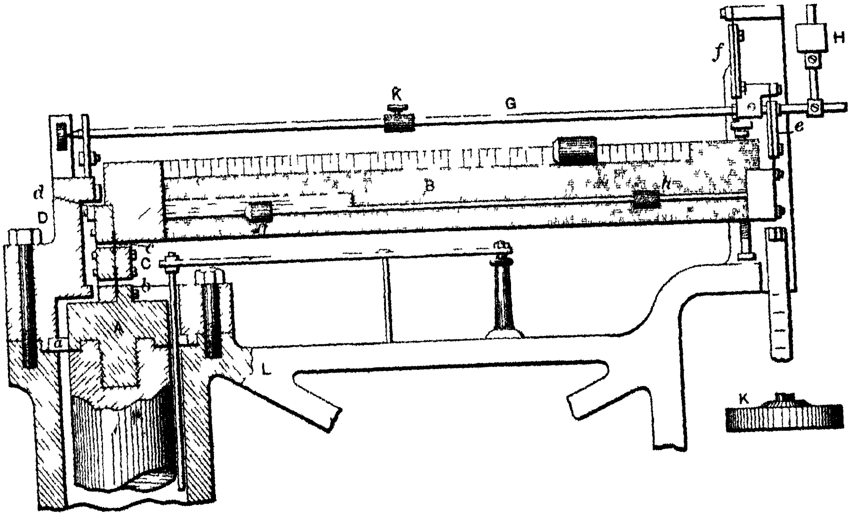

FIG. 61.—BEAM FOR PLATFORM-SCALE.

Machines have been built by this firm since 1876. The form of the first machine constructed was essentially that of a long weighing-beam suspended in a frame and connected by differential levers to the specimen, the power being applied by a hydraulic press. The latter forms are more compact. The standard hydraulic machine as constructed by this firm is shown in Fig. 63. In this machine the cylinder of the hydraulic press, which is situated directly beneath the specimen, is movable, and the piston is fixed.

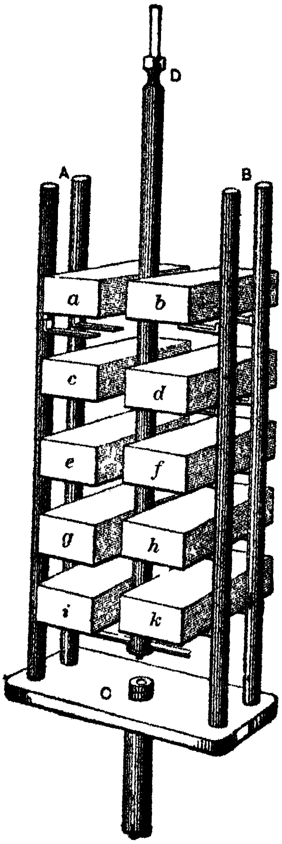

FIG. 62. — DEVICE FOR ADDING AND REMOVING WEIGHTS.

This motion is transmitted through the specimen, and is resisted by the weighing levers at the top of the machine, which are connected by rods and levers to the scale-frame. Two platforms connected by a frame are carried by the weighing levers: the upper one is slotted to receive the wedges for holding the specimen; the lower one forms a plane table. The intermediate platform, or draw-head, can be adjusted in different positions by turning the nuts on the screws shown in the cut. For tension-strains the specimen is placed between the upper and intermediate head; for compression it is placed between the intermediate and lower heads. An attachment is often added to the lower platform, so that transverse strains can be applied.

The hydraulic cylinder is connected by two screw rods to the intermediate platform or draw-head, and when it is forced downward by the operation of the pump this draw-head is moved in the same direction and at the same rate.

56. Riehlé Power Machines. — The machines in which power is applied by gearing are now more generally used than hydraulic

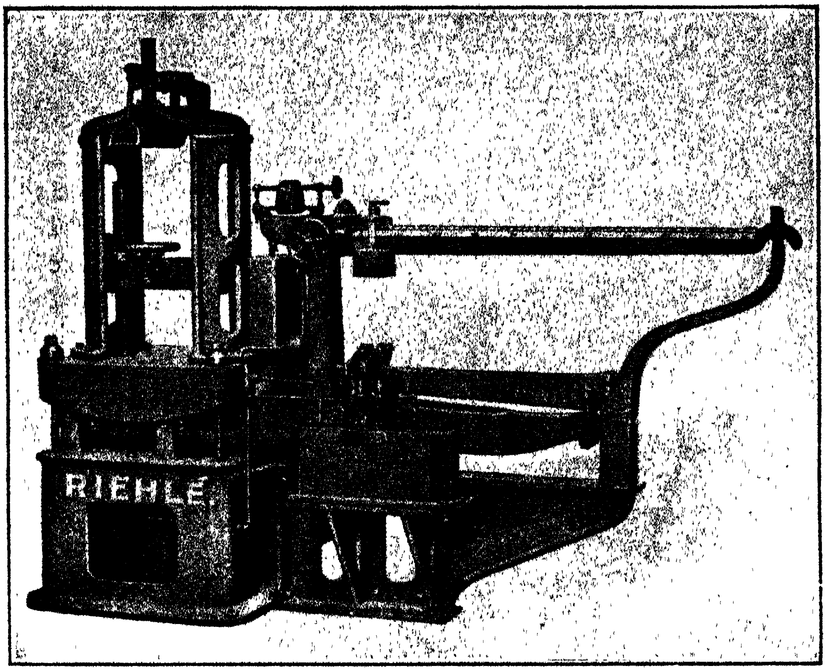

FIG. 63. — RIEHLÉ TESTING-MACHINE FOR TENSION, COMPRESSION, AND TRANSVERSE LOADING. FRONT VIEW.

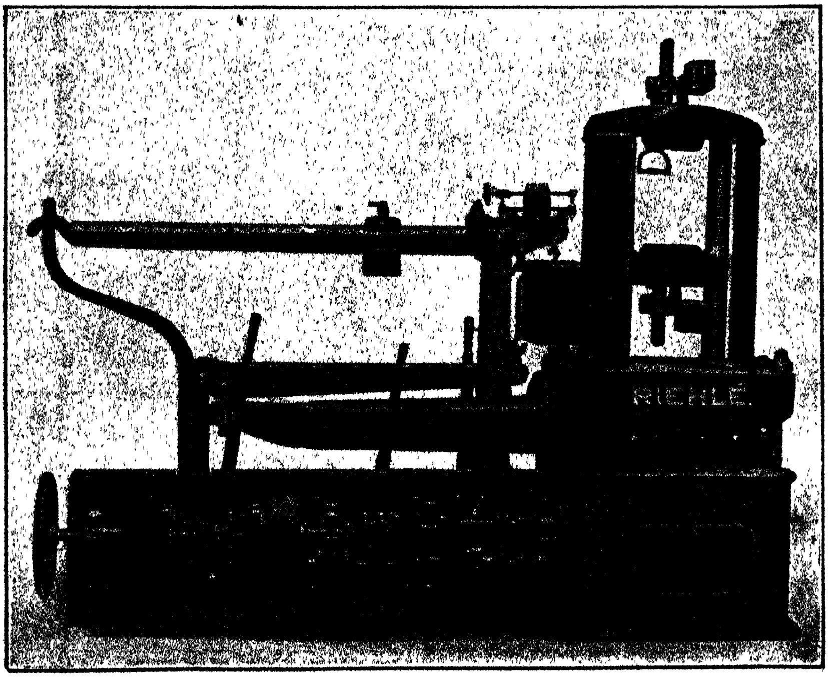

FIG. 64. — RIEHLÉ TESTING-MACHINE FOR TENSION, COMPRESSION, AND TRANSVERSE LOADING. BACK VIEW.

machines. Fig. 64 shows the design of geared machine now built by Riehlé Brothers. In this machine both the gearing for applying the power and the levers connected with the weighing apparatus are near the floor and below the specimen, thus giving the machine great stability. The heads for holding the specimen are arranged as in the hydraulic machine, and power is applied to move the intermediate platform up or down as required. The upper head and lower platform form a part of the weighing system. The intermediate or draw-head may be moved either by friction-wheels or spur-gears at various speeds, which are regulated by two levers convenient to the operator standing near the scale-beam.

The poise can be moved backward or forward on the scale-beam, without disturbing the balance, by means of a hand-wheel, opposite the fulcrum on which the scale-beam rests.

The scale-beam can be read to minute divisions by a vernier on the poise.

57. **Olsen Testing-machine.** — The machines of Tinius Olsen & Co. of Philadelphia are all operated by gearing, driven by hand in the machines of small capacity, and by power in those of larger capacity.

The general form of the machine is shown in Fig. 65, from which it is seen that the principles of construction are the same as in the machine last described.

The intermediate platform or draw-head is operated by four screws instead of by two, and there is a marked difference in the arrangement of the weighing-levers and in the gearing.

The machine can be operated at various rates of speed in either direction, and is readily controlled by convenient levers.

58. **Thurston's Torsion Testing-machine.** — Both the breaking-strength and the modulus of rigidity can be obtained from the autographic testing-machine invented by Professor Thurston in 1872.

In this machine, Fig. 66, the power is applied by a crank at one side, tending to rotate the specimen, the specimen being connected at the opposite end to a pendulum with a heavy weight.

The resistance offered by the pendulum is the measure of the force applied, since it is equal to the length of the lever-arm into the sine of the angle of inclination, multiplied by the constant weight P,

Fig. 67. A pencil is carried in the axis of the pendulum produced, and at the same time is moved parallel to the axis of the test-piece by a guide curved in proportion to the sine of the angle of deviation of the pendulum, so that the pencil moves in the direction of the axis of the specimen an amount proportional to the sine of this angle. A drum carrying a sheet of paper is moved at the same rate as the end of the specimen to which the power is applied. Now if the pencil be made to trace a line, it will move a distance around the drum which is equal to the angle of torsion (α) expressed in degrees or π measure, and it will move a distance parallel to the

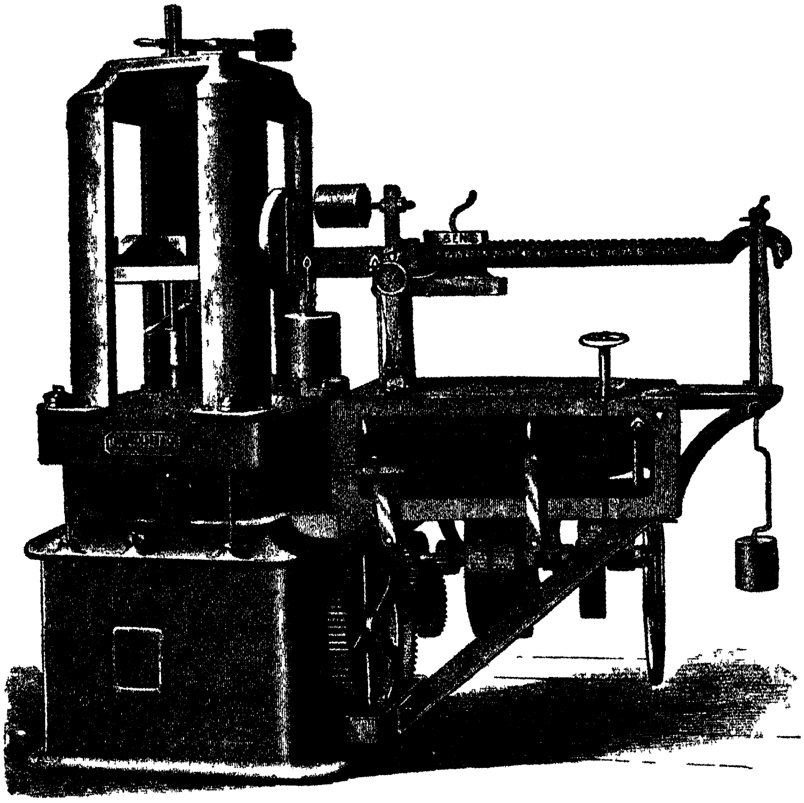

Fig 65.—The Olsen Testing-machine for Tension, Compression, and Transverse Loading.

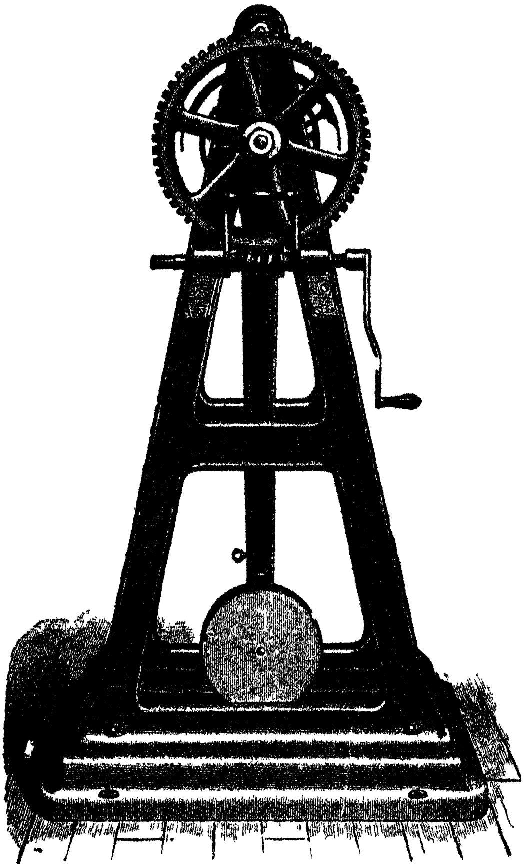

FIG 66 — THURSTON'S TORSION TESTING-MACHINE.

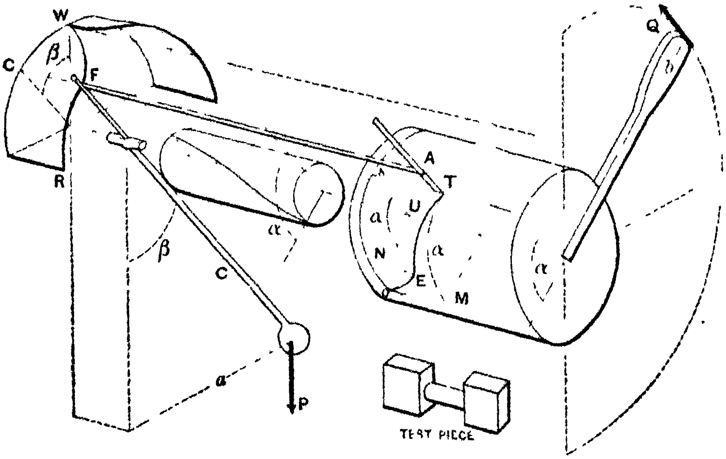

FIG. 67.

axis of the test-piece proportional to the moment of external forces, Pa.

The diagram, Fig. 67, from Church's "Mechanics of Engineering," shows the working portions of the machine very clearly. In the figure, P is the pendulum, the upper end of which moves past the guide WR, and is connected by the link FA with the pencil AT. The diagram is drawn on a sheet of paper on the drum, which is rotated by the lever b. The drum moves through the angle α, relatively to the pendulum which moves through the angle β. The test-piece is inserted between the pendulum and drum.

The value of α in degrees can be found by dividing the distance on the diagram by the length of one degree on the surface of the paper on the drum, which may be found by measurement and calculation.

Application of the Equations to the Strain-diagram. —For the breaking-load, equation (16) of Chapter III, may be written.

$$M = \frac{sI_p}{e}.$$

The external moment M equals $Pr \sin \beta$, in which P is the fixed weight, r the length of the pendulum, β the angle made with the vertical. Hence

$$Pr \sin \beta = sI_p \div e.$$

In this equation P and r are constant, and depend upon the machine; I_p and e are constant, and depend upon the test-piece. Sin β is the ordinate in inches on the autographic strain-diagram, and can be measured; knowing the constant, s may be computed from

$$s = Pre \sin \beta \div I_p.$$

For the modulus of rigidity, apply equation (19), Chapter III, page 62.

$$E_s = \frac{s}{\delta}.$$

The modulus of resilience (see equation (18), page 62) is the area of the diagram within the elastic limit, expressed in absolute units.

$$U_s = \tfrac{1}{2} s'\delta' \text{ for unit of material.}$$

The Helix Angle $\delta = e\alpha \div l$, in which l is the length of the specimen in inches.

Machine Constants. — To Obtain the Constants of the Machine. — First, the external moment Pa. This is obtained on the principle that it is equal to any other external moment which holds it in equilibrium. Swing the pendulum until its centre-line is horizontal; support it in this position by a strut resting on a pair of scales; the product of the corrected reading of the scales into the distance to the axis on the arm will give M. Check this result by trials with the strut at different points. Second, the value of the scale of ordinates can be obtained by measuring the ordinate for $\beta = 90°$ and for $\beta = 30°$, since sine $90° = 1$ and sine $30° = \frac{1}{2}$. Third, the value of the scale of abscissæ can be obtained by dividing the abscissa on the diagram by the circumference of the drum including the paper. This may be expressed in degrees by considering the circumference $= 360°$.

Constants of the Material are obtained by measuring the dimensions of the specimen. The values of I_p and e are given on page 62.

Conditions of Accuracy. — In obtaining these values, the following conditions are assumed: Firstly, the test-piece is exactly in the center of motion of the pendulum and of the drum; secondly, the pencil is in line of the pendulum produced; thirdly, the curve of the guides is that of the sine of the angle of deviation; and, fourthly, the specimen is held firmly from rotation by the shackles or wedges, and yet allowed longitudinal motion. These constitute the adjustments of the machine, and must be carefully examined before each test. Any eccentricity of the axis of the specimen will lead to serious error.

59. Power Torsion-machines.—The *Riehlé power torsion-machine* is shown in Fig. 68. Power is applied at various rates of speed by means of the gearing shown. The specimen is held by means of two chucks: the one on the left is rotated by the power applied; the one on the right is prevented from rotating by a system of levers, so connected to the scale-beam that when it is balanced the reading is proportional to the torsional force or external moment transmitted through the specimen, expressed in foot-pounds, inch-pounds, or any other units desired. The weighing head is suspended so as to permit free elongation of the specimen. The chucks used have self-centering jaws which will hold the specimen rigidly and central during application of the stress. In the Riehlé machine shown, the adjustment for specimens of various lengths is made by moving the weighing head.

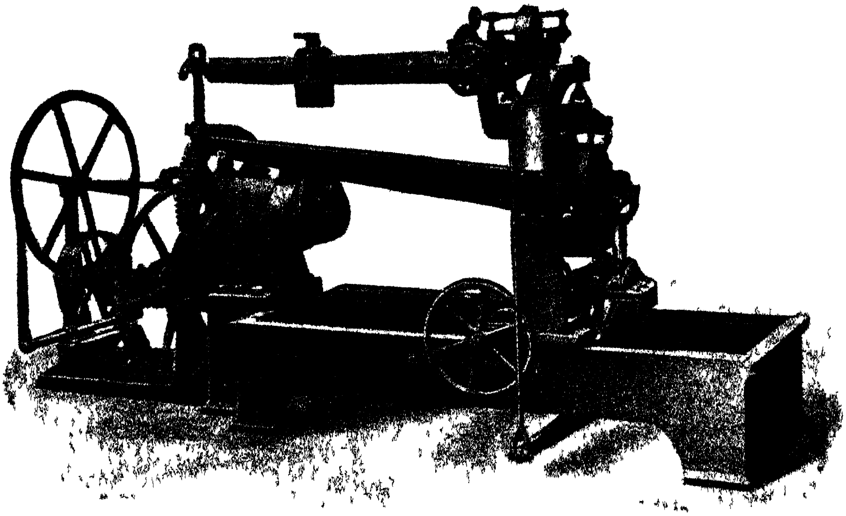

FIG. 68.—RIEHLÉ POWER TORSION TESTING MACHINE.

It is in general not best to determine the angle of torsion from a graduated scale on the movable chuck unless the slip of the specimen in the jaws is corrected for. The Riehlé Co. make a torsion indicator, which is applied directly to the specimen and thus obviates any inaccuracy due to slip. The construction of this indicator is shown in Figs. 69 and 70 and needs no further explanation. Two of these are used, clamped to the specimen any desired distance apart. The reading of the dial near the weighing head is subtracted from that near the power head.

Fig. 71 shows the *Olsen power torsion machine*. Here the movable or power chuck is shown at the right, the stationary or weighing chuck at the left. In the particular machine shown, that at Sibley College, the scale beam shows the force in pounds acting at an arm of 20″.

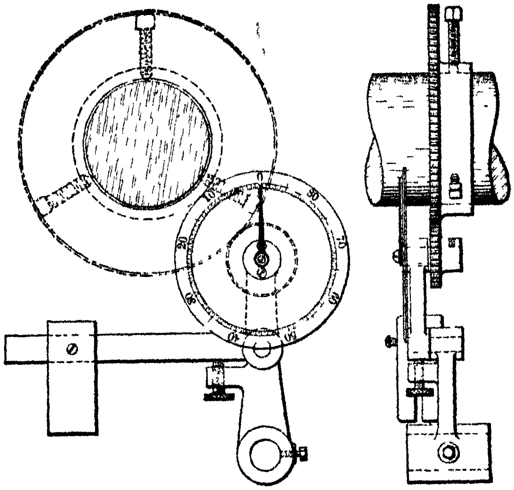

Fig 69 Fig 70

Riehlé Torsion Meter.

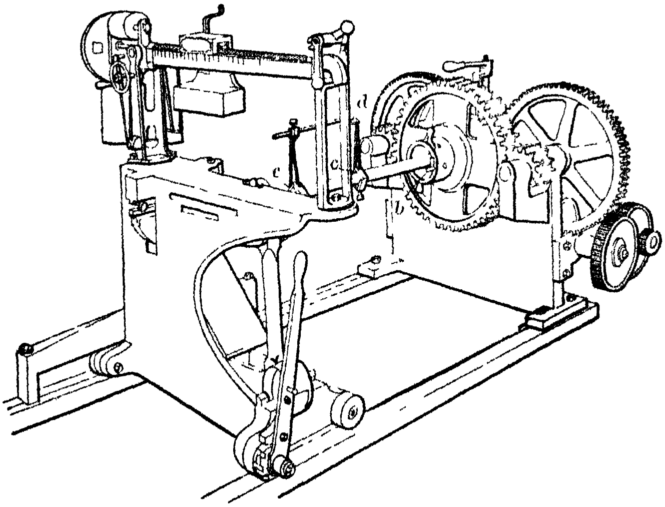

Fig. 71. — Olsen Torsion Testing-machine.

In the Olsen machine the angle of torsion may be measured by clamping dogs on the specimen at each end so as to engage the projections, shown at *b*, Fig. 71, of the index-rings, which are free to move over the graduated scales of the chucks. The angle of torsion of the specimen, for a length represented by the distance between the centers of the dogs, is the angle turned through by the movable chuck less the sum of the angles through which the index-rings are pushed by the dogs. Let α_1 = angle through which movable chuck is rotated, α_2 = angle through which index-ring on the movable chuck is pushed by the dog, α_3 = angle through which index-ring on fixed chuck is pushed by the dog, and α = angle of torsion. Then

$$\alpha = \alpha_1 - (\alpha_2 + \alpha_3).$$

This angle may also be measured through short ranges by means of two index-arms clamped to the specimen, as shown at *c*, Fig. 71. One arm carries a pointer which plays over an arc (*d*), graduated in inches, whose center of curvature is the center of the specimen. The distance traversed by the pointer divided by the radius of the arc gives the angle of torsion in circular measure.

The constant of the Olsen machine, or the value of the graduations on the scale-beam, may be found as follows (see Fig. 72): The fixed chuck is rigidly connected to link *K* as shown. The torsion moment (*Pa*) on the specimen tends to rotate the chuck and link as indicated by the arrow. The only additional forces acting on *K* are the vertical forces of strut P_1 and of the frame through the knife-edges at *R*. The right end of link *K* is prevented from dropping down, when no load is on the specimen, by a strut acting upward at *R* (not shown in figure). *R* may therefore act either upward or downward, depending upon the intensity of *Pa*. The weight of *K* may, however, be entirely neglected, since the counterpoise of the machine may be so set that the system is in equilibrium with no stress on the specimen.

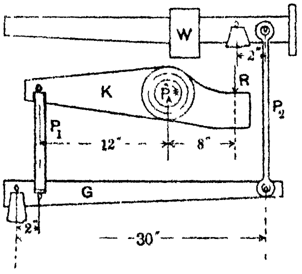

FIG. 72. — WEIGHING HEAD OF OLSEN TORSION-MACHINE.

With the dimensions shown, weight of poise = 40 pounds, length between divisions on scale-beam = $\frac{2}{3}$ inch, consider K as a free body. Then $\Sigma(Pa) = 0$ and $\Sigma Y = 0$. From which

$$Pa = 12\,P_1 + 8\,R \text{ and } P_1 = R,$$

or
$$Pa = 20\,P_1 \tag{1}$$

P_1 acts at a lever-arm of 2 inches on the lower lever G, and P_2 acts at a lever-arm of 30 inches. Then

$$2\,P_1 = 30\,P_2 \text{ and } P_1 = 15\,P_2 \tag{2}$$

P_2 acts on scale-beam at a lever-arm of 2 inches, and this moment must be balanced by moving the poise W along a distance x. From which

$$2\,P_2 = Wx \tag{3}$$

From (1), (2), and (3) we have

$$Pa = 20 \times 15 \times 20\,x.$$

Make $x = 1$ scale division $= \frac{2}{3}$ inch, then

$$Pa = 4000 \text{ inch-pounds.}$$

Since the value of each division as marked on scale-beam is 200, the constant of the machine is 20.

For an accurate determination of the angle of torsion, it is important that the specimen be kept straight during the application of stress, and that the angle of torsion be measured from arcs or scales having the same center as the specimen.

60. Impact-testing Machine. — *The Drop Test — Testing by Impact.* — This test is recommended for material used in machinery, railroad construction, and generally whenever the material is likely to receive shocks or blows in use.

It is usually performed by letting a heavy weight fall on to the material to be tested. The Committee on Standard Tests of the American Society of Mechanical Engineers *recommend that the* standard machine for this purpose consist of a gallows or framework operating a drop of twenty feet, the weight to be 2000 pounds, the machine to be arranged substantially like a pile-driver. The impact machine designed by Mr. Heisler, Fig. 73, consists of a pendulum with a heavy bob, which delivers a blow on the center of a bar securely held on two knife-edge supports affixed to a heavy mass of metal. This machine is especially designed for comparative tests of cast iron; it is furnished with an arc graduated to read the vertical fall of the bob in feet, and a trip device for dropping the ram from any point in the arc. A paper drum can be arranged for automatically recording the deflection of the test-pieces.

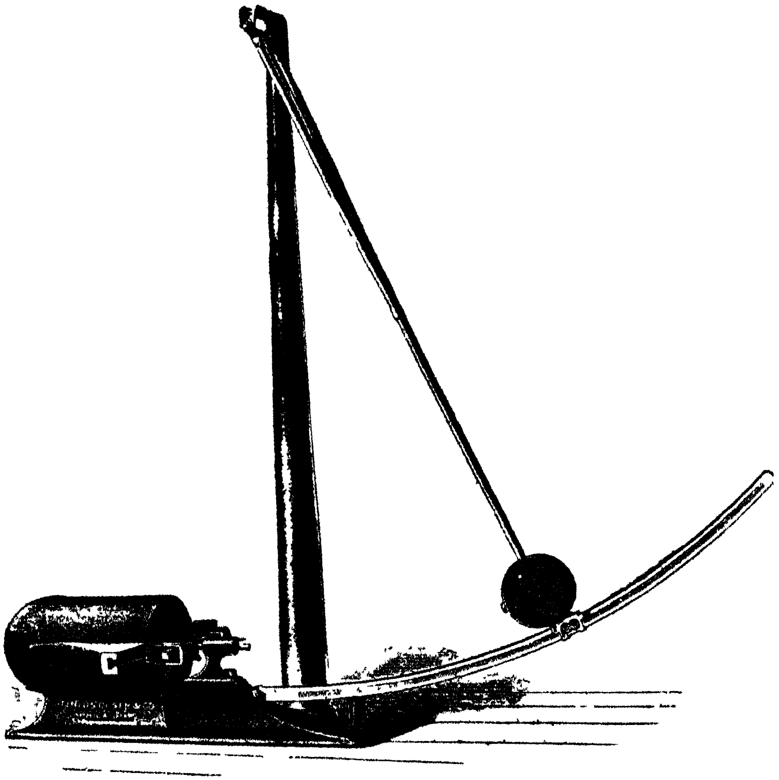

FIG. 73.—HEISLER IMPACT TESTING-MACHINE.

Let W = the weight of the bob;
h = the distance fallen through;
P = centre-load;
λ = deflection.

Then

$$Wh = \tfrac{1}{2} P\lambda.$$

Hence,

$$P = 2\,Wh \div \lambda.$$

61. Machines for Testing Cement. — Cement mortar can be formed into cubes, and after hardening can be tested in the usual testing-machines for compression; but tensile tests are usually required, and for this purpose a delicate machine with special shackles is needed. In order that the tests may give correct results, it is necessary that the power be applied uniformly, and absolutely in the line of the axis of the specimen; and to make different tests comparable, the specimen, or, as it is called, the briquette, must be always of the same shape and size, and made in exactly the same manner. The engraving (Fig. 74) shows *Fairbanks' Automatic Cement-tester*, in which the power is applied by the dropping of shot into the pail *F*. The specimen is held between clamps, which are regulated at the proper distance apart by the screw *P*. At the instant of rupture the scale-beam *D* falls, closes a valve, and stops the flow of shot. In Fig. 74 *S* is a closed mould for forming a briquette, and *U* a briquette ready for test.

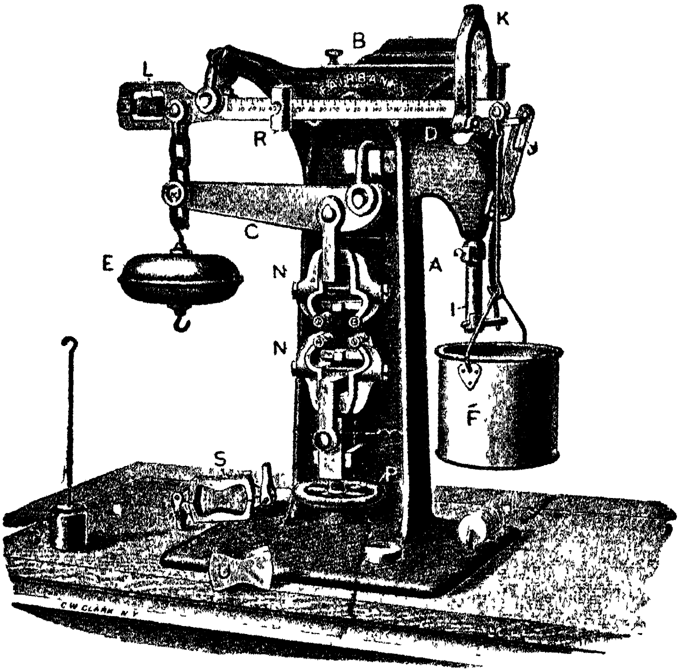

FIG. 74. — FAIRBANKS' CEMENT-TESTING MACHINE.

Directions. — Hang the cup *F* on the end of the beam *D*, as shown in the illustration. See that the poise *R* is at the zero-mark, and balance the beam by turning the counter weight *L*.

Place the shot in the hopper *B*, place the specimen in the clamps *NN*, and adjust the hand-wheel *P* so that the graduated beam *D* will rise nearly to the stop *K*. Open the automatic valve *I* so as to allow the shot to run slowly. Stand back and let the machine make the test.

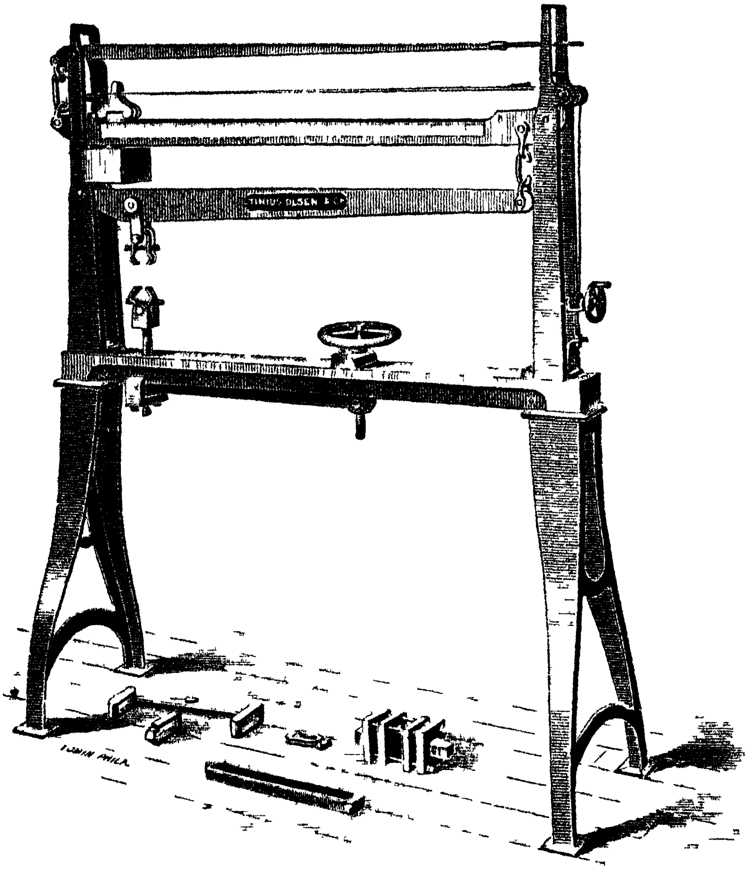

Fig. 75. — Olsen's Cement Testing Machine.

When the specimen breaks, the beam *D* drops and closes the valve *I*. Remove the cup with the shot in it, and hang the counterpoise-weight *G* in its place. Hang the cup *F* on the hook under the large balance-ball *E*, and proceed to weigh the shot in the ordinary way, using the poise *R* on the graduated beam *D* and the weights *H* on the counterpoise weight *G*. The result will show the number of pounds required to break the specimen.

Owing, apparently, to a certain give in the shackles and lever, it sometimes becomes necessary to turn the wheel *P* to prevent *D* from striking the lower stop before the test is complete. Under high loads it becomes difficult to do this uniformly by hand. In the later forms of this machine an attachment is now located in the base of the machine whereby *P* can be turned through gearing, making the action more uniform and less apt to affect the strength of the specimen by jars.

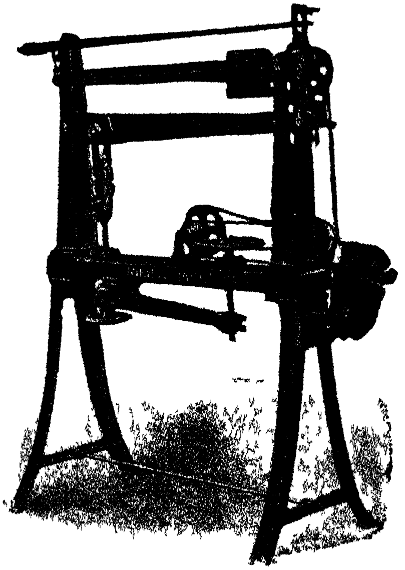

Fig. 76.—Riehlé Cement-testing Machine.

Automatic Machines, similar in their action to the Fairbanks, are also built by Tinius Olsen & Co., and by Riehlé Bros. Testing Machine Co., both of Philadelphia. These firms also make cement-testing machines of a different type, of which Figs. 75 and 76 are examples.

The *Olsen Cement-tester* is shown in Fig. 75. The power is applied by the hand-wheel and screw, so that it strains the briquette very slowly. The poise on the scale-beam is moved by turning a crank so that the beam can readily be kept floating.

The *Riehlé Cement-tester* is shown in Fig. 76. The briquette to be tested is placed between two shackles mounted on pivots so as to be free to turn in every direction.

Power is applied to the specimen through belts and gearing as shown, and is measured by the reading on the scale-beam at the position of the poise.

TESTING-MACHINE ACCESSORIES.

62. General Requirements of Instruments for Measuring Strains.— In the testing of materials it is necessary to measure the amount of strain or distortion of the body, in order to compute ductility, modulus of elasticity, etc. The ductility or percentage of ultimate deformation, since the latter is usually a large quantity, can often be obtained by measurement with ordinary scales and calipers. Thus, in the tension-test of a steel bar 8 inches long, it will increase in length before rupture nearly or quite 2 inches; if in the measure of this quantity an error equal to one-fiftieth of an inch be made, the resulting error in ductility is only one-half of one per cent. In the measurement of deformation occurring within the elastic limit the case is very different, as the deformation is very small, and consequently a very small error is sufficient to make a great percentage difference in the result.

The instruments used for the purpose of measuring elongation are called *extensometers*, and vary greatly in form and in principle of construction. The instrument is generally attached to the test-piece, either on one or on both sides, and the deformation is obtained by direct measurement with one or two micrometer-screws, or by the use of levers which multiply the deformation so that the results can be read on an ordinary scale. As a rule, instruments which attach to one side of the test-piece will give erroneous readings if the test-piece either be initially curved, or deformed so as to draw its axis out of a right line, and this error may be large or small, as the conditions vary.

The extensometers in use generally consist of some form of multiplying-lever the free end of which moves over a scale which may or may not be provided with a vernier, or of a micrometer-screw, which is used to measure the distance between fixed points attached to the specimen or of some form of mirror apparatus, or various forms of cathetometers.

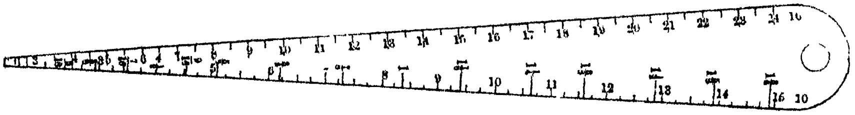

FIG. 77. — WEDGE SCALE.

63. Various Forms of Extensometers. *The Wedge Scale.* — The wedge-shaped scale, Fig. 77, which could be crowded between two fixed points on the test-piece, was one of the earliest devices to be used. In using the scale two projecting points were attached to the specimen, and as these points separated, the scale could be inserted farther, and the distance measured.

The Bauschinger Roller and Mirror Extensometer. — To Professor Bauschinger belongs the credit of first systematically taking double measurements on opposite sides of a test-bar. The general principle of his apparatus is shown in Fig. 78. It is seen to consist of two knife-edged clips, *b*, *b*, which are connected to the specimen and carry two hard ebonite rollers, *d*, *d*, which turn on accurately centered spindles. The spindles are prolonged, and support mirrors,

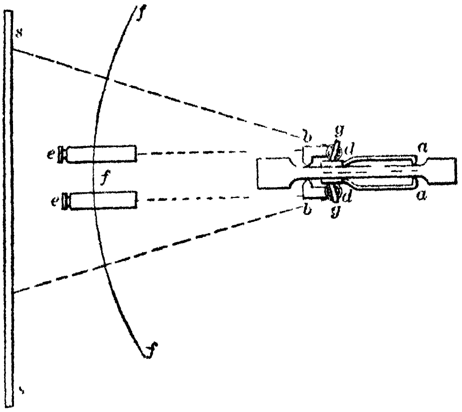

FIG 78. — BAUSCHINGER'S MIRROR APPARATUS.

g, *g*, which rotate in the plane of the figure as the spindles rotate. Clips, *a*, *a*, are fastened to each side of the test-piece at the opposite extremity, and are connected by spring-pieces, with the rollers. The spring-pieces are slightly roughened by file, and turn the rollers by frictional contact, so that the least extension of the test-piece causes a rotation of the mirror through an angle. If a scale be placed at *s*, *s*, and telescopes at *e*, *e*, the reflection of the scale will be seen in the mirror in looking through the telescope, and any extension of the test-piece will cause a variation in the reading of the scale as seen in the mirror. The apparatus is equivalent to a lever apparatus having for a small arm the radius of the roller *g*, and for a long arm the double distance of the scale from the mirror. With this instrument it is evidently possible to obtain very accurate measurements, but on the other hand the instrument is very cumbrous and difficult to use. The mean of the two readings with the Bauschinger instrument is the true extension of the piece.

Professor Unwin obviates the use of two mirrors and two telescopes by attaching clips to the center of the specimen and having the single mirror revolve in a plane at right angles with the plane passing through the clips and the axis of the specimen.

Strohmeyer's Roller Extensometer, Fig. 79, was designed in 1886, and is a double-roller extensometer. The apparatus consists of a roller carrying a needle, which is centered with respect to a graduated scale. The roller moves between side-bars extending to clips which are fastened to each end of the specimen. The tension between these side-bars can be regulated by a spring with a screw adjustment. The objections to this form of extensometer are due, first, to slipping of side-bars on the roller, and second, to the difficulty in making the roller perfectly round.

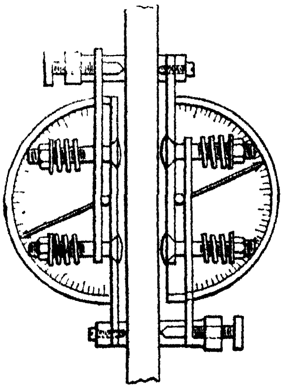

FIG. 79. — THE STROHMEYER EXTENSOMETER.

Regarding the various forms of extensometers, the writer would say that his experience has covered the use of nearly every form mentioned, and none have proved to be superior in accuracy to that with the double micrometer-screw, and few can be applied so readily.

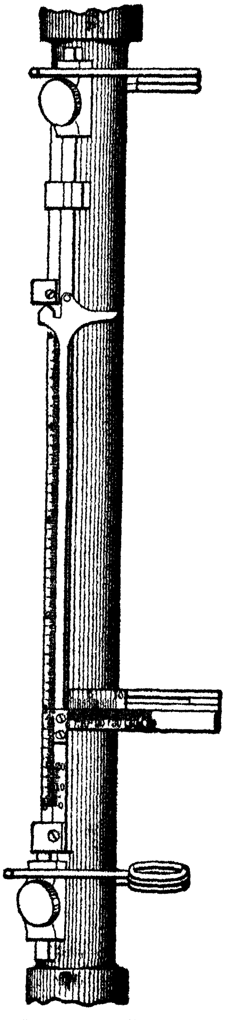

FIG. 80.—PAINE EXTENSOMETER.

The Paine Extensometer.—This instrument, shown in Fig. 80, operates on the principle of the bell-crank lever, the long arm moving a vernier over a scale at right angles to the axis of the specimen. It reads by the scale to thousandths of an inch, and by means of the vernier to one ten-thousandth of an inch. Points on the instrument are fitted to indentations in one side of the test-piece, and the instrument is held in place by spring clips. It is of historical importance, having been invented by Colonel W. H. Paine, and used in the tests of material for the Brooklyn Bridge, and also on the cables of the Niagara Suspension Bridge when, a few years since, the question of its strength was under investigation.

Buzby Hair-line Extensometer.—This is an extensometer in which the deformation is utilized to rotate a small friction-roller connected with a graduated disk as shown in Fig. 81. A projecting pin placed in the axis of the graduated disk is held between two parallel bars, each of which is connected to the specimen. The deformation is magnified an amount proportional to the ratio of diameters of the disk and pin. The amount of deformation is read by noting the number of subdivisions of the disk passing the hair-line. To prevent error of parallax in reading, a small mirror is placed back of the graduations, and readings are to be taken when the graduations, the cross-hair, and its reflection are in line. In the late styles of this instrument the disk is made of aluminum, with open spokes, to reduce its weight.

To operate this instrument it is only necessary to clamp it to the specimen, to adjust the mirror and cross-hair, and then to revolve the disk by hand until the zero-line corresponds with the cross-hair and its reflection. Stress is then applied to the specimen, and readings taken as desired in the manner described.

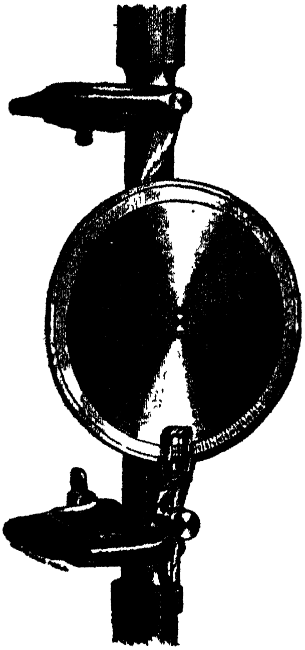

FIG. 81. — BUZBY HAIR-LINE EXTENSOMETER.

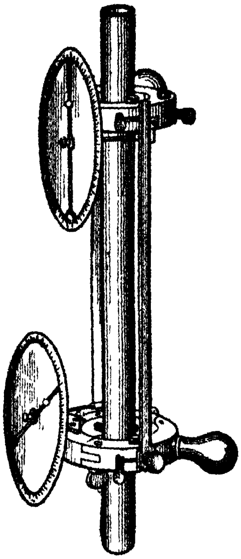

FIG. 82. — JOHNSON EXTENSOMETER.

Johnson Extensometer. — Johnson extensometer, shown in Fig. 82, is a modification of the Strohmeyer, the elongation being denoted by the motion of a needle over a graduated scale. The elongation for each side is shown separately, and the algebraic sum of the two readings gives the total elongation.

Thurston's Extensometer. — This extensometer was designed by Prof. R. H. Thurston and Mr. Wm. Kent, and was the first to employ two micrometer-screws, at equal distances from the axis of the specimen. These were connected to a battery and an electric bell in such a manner that the contact of the micrometer-screws was indicated by sound of the bell. The method of using this instrument is essentially the same as that of the Henning and Marshall instrument, to be described later.

With instruments of this nature a slight bending in the specimen will be corrected by taking the average of the two readings.

The accuracy of such extensometers depends on —

1. The accuracy of the micrometer-screws.
2. The screws to be compensating must be two in number, in the same plane, and at equal distances from the axis of the specimen.
3. The framework and clamping device must hold the micrometers rigidly in place, and yet not interfere with the application of stress.

The Henning Extensometer. — This instrument, which was designed by G. C. Henning and C. A. Marshall, is shown in Fig. 83. It is constructed on the same general principles as the Thurston Extensometer, but the clamps which are attached to the specimen are heavier, and are made so that they are held firmly in position by springs up to the instant of rupture. This extensometer is furnished with links connecting the two parts together. The links are used to hold the heads exactly eight inches apart, and are unhooked from the upper head before load is applied to the specimen. The micrometer is connected to an electric bell in the same manner as the Thurston extensometer.

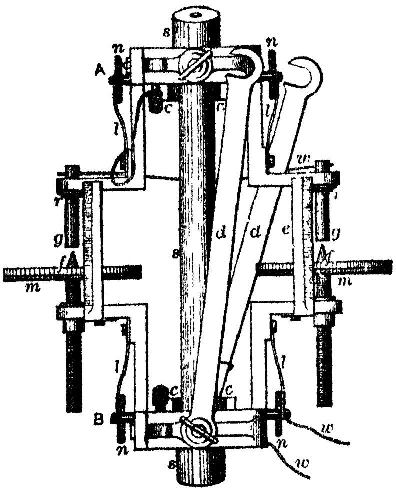

FIG. 83. — HENNING EXTENSOMETER.

*Henning's Mirror Extensometer.** — In 1896 Gus. C. Henning designed a mirror extensometer differing in several particulars from that of Bauschinger. The instrument is intended for accurate measurements of the extension or compression on both sides of the test-piece within the elastic limit, and is said to fulfil the following conditions: (*a*) It is applicable for measures of extension or compression. (*b*) Readings in either direction, negative or positive, can be taken without interruption or adjustment. (*c*) The instrument is free from changes of shape during the test. (*d*) There is neither slip nor play of the working parts.

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* See Transactions American Society Mechanical Engineers, Vol. XVIII.

The instrument consists of two parts; the first is a telescope provided with levelling-screws, mounted on a horizontal and vertical axis and furnished with supports for two linear scales, which may be arranged so that the reflection will show in mirrors attached to the specimen. The second part consists of a frame which can be fastened to the test-specimen near one end by opposing, pointed screws, and which is connected to spindles carrying the mirrors by spring side-bars. A portion of each mirror-spindle is double knife-edged, and when adjusted is brought in contact on one side with the test-piece, and on the other with the spring side-bar. The elongation of the test-piece causes an angular motion of the mirror, which in turn causes a multiplied motion of the reflection of the scale as seen from the telescope. The mirrors are so arranged that the reflections from both scales can be seen continually and without adjustment of the telescope, and the apparatus as a whole has fewer parts and is more readily adjusted than the Bauschinger. It is limited to a total elongation of about 0.04 inch and hence is accurate only for measurements within the elastic limit.

The Marshall Extensometer. — This extensometer, shown in Fig. 84, is the latest design of the late Mr. C. A. Marshall. Its principal difference from the Thurston extensometer is in the convenient form of clamps, which are well shown in the cut, and in the spring apparatus for steadying the lower part.

The micrometer-screw used with this instrument has a motion of only one inch. When the motion exceeds the range of the micrometer-screws, the movable bars *BP*, *B'P'* are changed in position, and a new series of readings can be taken with the micrometer-screw.

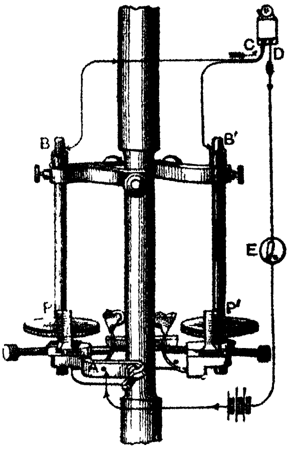


FIG. 84. — THE MARSHALL EXTENSOMETER.

The following are the directions for connecting up the electric bell circuits used with extensometers of this type:

Run wire (Fig. 84) from one terminal of battery to lower clamp at *A*, from *B* and *B'* to binding-post *C* on the electric bell, from the other binding-post marked *D* to switch *E*, and from there back to the other terminal of battery.

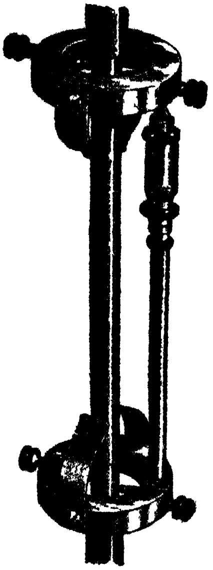


FIG 85. — BOSTON EXTENSOMETER

To measure deformation, screw up the micrometer screw on one side until the bell indicates contact, then back off very slowly until the circuit breaks, then take the reading. After this, back off the same screw a little more to prevent its making contact when working with the other side, and repeat the same process with the other screw.

Boston Micrometer Extensometer.—This instrument consists, as shown in Fig. 85, of the graduated micrometer-screw, reading in thousandths up to one inch, and having pointed extension-pieces attached, for gauging the distance between the small projections on the collars fastened to the specimen at the proper distance. These collars are made partly self-adjusting by the springs which help to centralize them. They are then clamped in place by means of the pointed set-screws on the sides, and measurements are made between the projections on opposite sides of the specimen and compared, to denote any changes in shape or variations in the two sides.

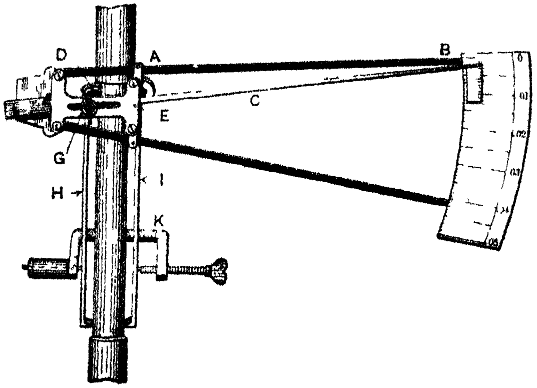


FIG. 86. — OLSEN NEW EXTENSOMETER.

The Brown and Sharpe micrometer can readily be used with similar collars, thus forming an extensometer; the accuracy of this form is considerably less than those in which the micrometers are fixed, but it will, however, be found with careful handling to give good results.

The New Extensometer. — This is a type which is finding extended use. Fig. 86 shows its construction, method of attachment and of operation without further explanation. By the use of auxiliary sleeves, the same type of instrument may also be used in the testing of wire, Fig. 87.

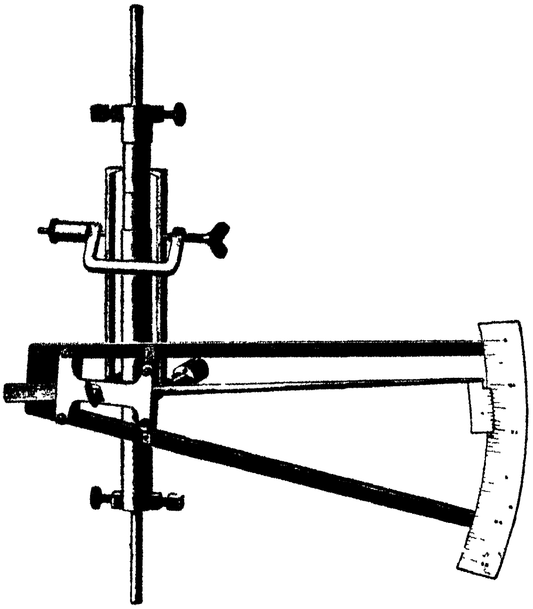


FIG. 87. — OLSEN NEW WIRE EXTENSOMETER.

Of the various extensometers described, the Paine, Buzby, and Marshall, are manufactured by Riehlé Bros., Philadelphia; the Thurston and New by Olsen of Philadelphia; the others, by the respective designers.

64. Combined Extensometer and Autographic Apparatus.— Various methods have been devised whereby the stress and strain diagram may be obtained directly by action of the testing machine. All of these devices consist essentially of a drum which carries the paper upon which the diagram is to be drawn and whose movement about its own axis is usually determined by the deformation of the test-piece multiplied in some ratio. Thus motion around the drum records strain. The pencil drawing the diagram is moved along the drum in some proportion to the movement of the poise along the weighing beam. Thus the combined motion of drum and of pencil produces the diagram.

Some of this autographic apparatus is quite elaborate. The Riehlé Gray apparatus built by Riehlé Bros. is able to produce a double diagram, one showing the entire curve to the breaking point of the specimen; the other, by multiplying the elongation within the elastic limit, shows the characteristics of the material up to the yield point. The type of diagram obtained is shown in Fig. 88.

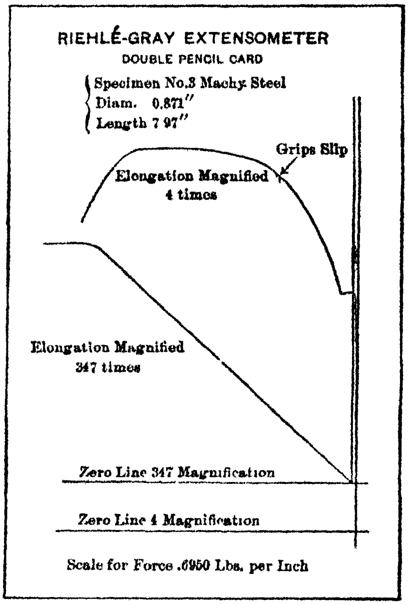


FIG. 88. — AUTOGRAPHIC RECORD

A simpler type of autographic apparatus, also built by the same company, is shown attached to the machine in Fig. 89. Here the deformation movement of the specimen is converted into circular motion of the drum, while the pencil moves up along the drum in proportion to the load.

The Olsen autographic device is shown attached to the machine in Fig. 90. The drum is here supported parallel to the beam. The pencil is carried by a fine threaded extension of the beam screw. Thus movement parallel to the drum represents load. The deformation of the specimen is followed by contact fingers by means of which the movement is multiplied five times and converted into the rotary motion of the drum. This mechanism is used to draw the entire diagram for the detection of the yield point, the determination of the breaking load, etc. For the more accurate work required within the elastic limit an auxiliary attachment is furnished which multiplies the deformation 500 times.

Concerning any autographic apparatus of this type the following points should be kept in mind. Owing to slip in the jaws in the case of tension and yielding of the supports in the case of compression, that part of the apparatus recording the deformation should never be attached to or rest against the heads of the machine. Instead of this, collars on the specimen should always be used. Further, the more elaborate the apparatus, the greater the

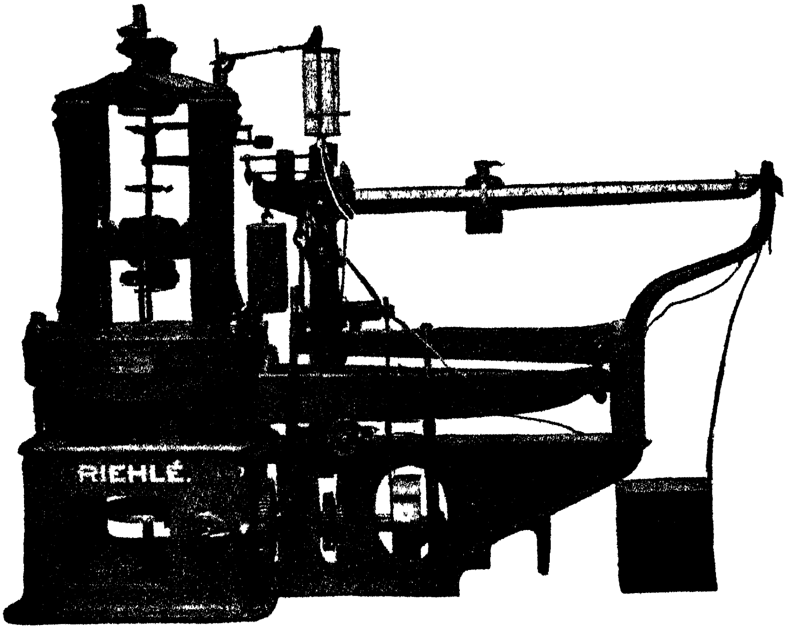


FIG. 89.—RIEHLÉ TESTING-MACHINE WITH AUTOGRAPHIC APPARATUS.

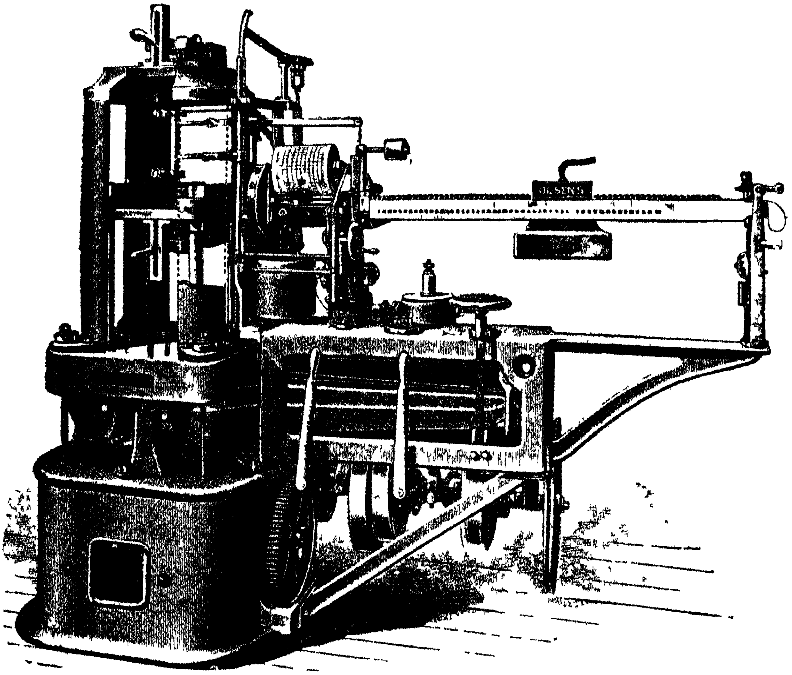


FIG. 90.—OLSEN TESTING-MACHINE WITH AUTOGRAPHIC APPARATUS

chance for error of adjustment, for lost motion in the bearings and for bending and yielding in the levers. All these things affect the result, and while errors thus induced may not seriously affect the determination of the yield point, maximum load, breaking load, and total elongation, the results for the modulus of elasticity as determined from the magnified curve should always be used with caution.

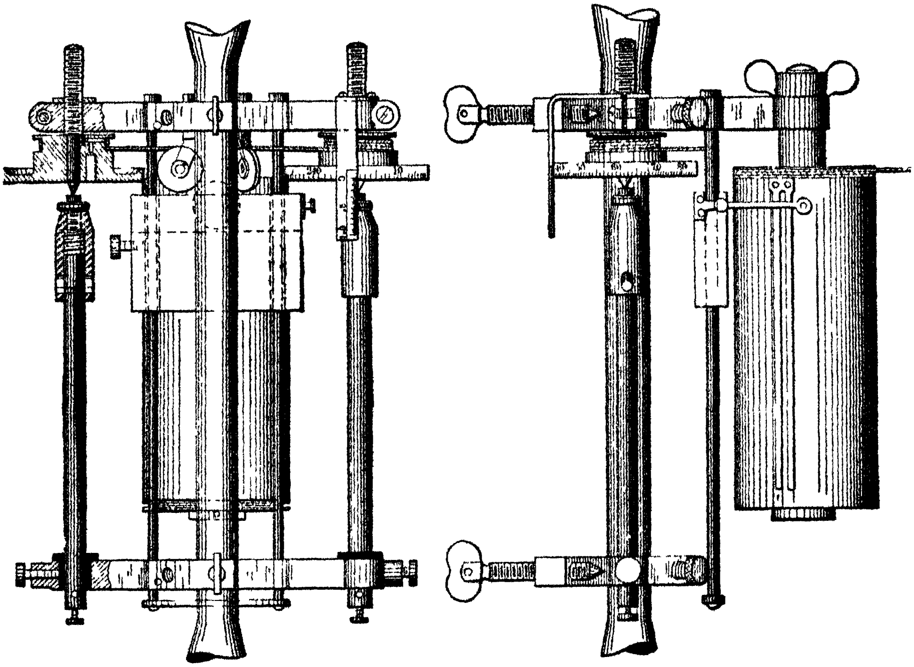


FIG. 91. — KENERSON AUTOGRAPHIC EXTENSOMETER.

The autographic apparatus so far described may be considered an integral part of the machine. Another type merely forms an auxiliary attachment to the ordinary extensometer. To this class belong the Henning pocket recorder and the Kenerson extensometer, the latter of which is shown in Fig. 91. In this instrument the upper clamp carries a drum whose motion about its axis is controlled by the movement of the poise on the weighing beam by means of a cord. The pencil is carried by a weight which can move up or down between two guide rods also carried by the upper clamp. This weight is suspended by means of two cords which pass over small guide sheaves and which are wrapped around two small pulleys carried by the micrometer screws. The rest of the apparatus is like an ordinary extensometer. As the specimen stretches and the contact points tend to separate, the weight pulling on the cords rotates the pulleys and maintains contact. Thus motion of the pencil downward is proportional to the deformation.

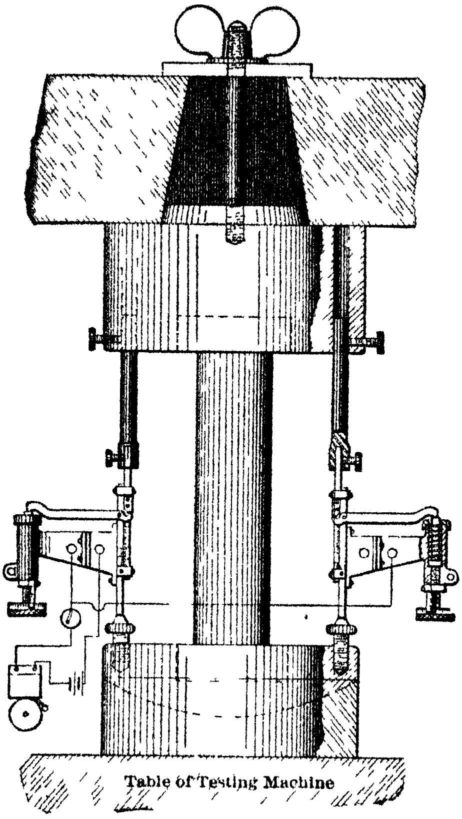


Fig. 92.—Riehlé Compression Micrometer.

65. Instruments for Measuring Torsion, Deflection, and Compression.—Instruments for measuring the angle of torsion have already been discussed under Torsion Machines, see Art. 58. Instruments for measuring the deflection of a specimen subjected to transverse stress are termed *deflectometers.*

The deflectometer usually used by the author consists of a light metal-frame of the same length as the test-piece, and arched or raised sufficiently in the center to hold a micrometer above the point to which measurements are to be taken. In using the deflectometer it is supported on the same bearings as the test-piece, and

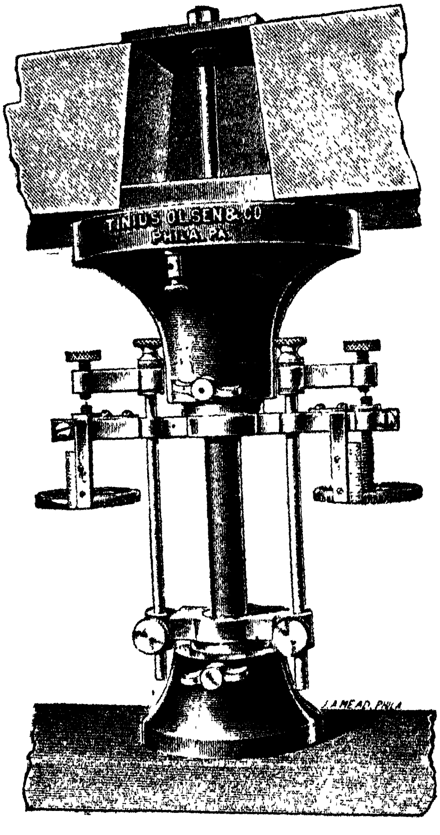


FIG. 93.—OLSEN COMPRESSION MICROMETER

measurements made to a point on the specimen or to a point on the testing-machine which moves downward as the specimen is deflected. This instrument eliminates any error of settlement in the supports. There are a number of different types of instruments which may be used to measure compression. Some of these are modifications of the extensometer idea, see Figs. 92 and 93. Both of these are

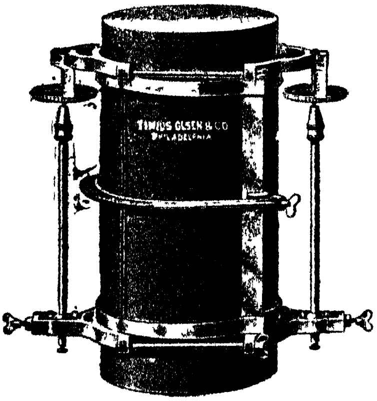


FIG. 94. — OLSEN COMPRESSION MICROMETER FOR CONCRETE, ETC.

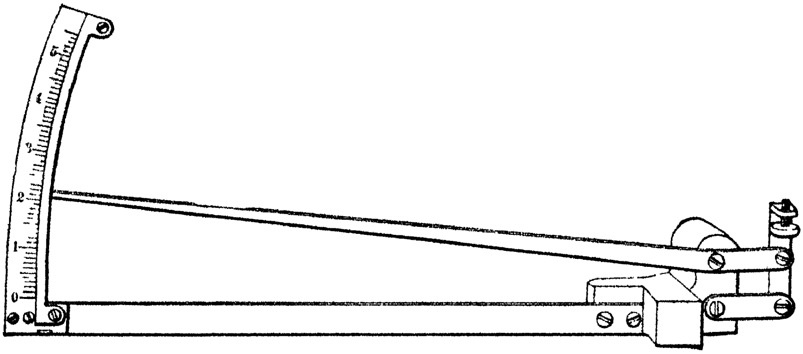


FIG. 95. — COMPRESSION MICROMETER OR DEFLECTOMETER.

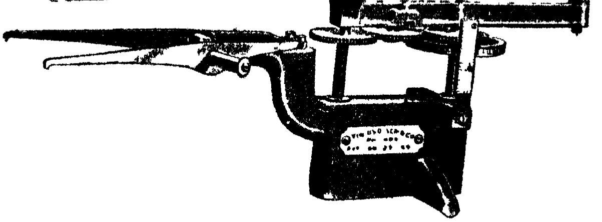
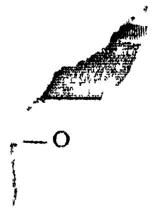


FIG. 96. — OLSEN COMPRESSION MICROMETER.

apparently open to the objection that they do not allow for the settling of the supports. The instrument shown attached to the cylindrical specimen in Fig. 94 avoids this source of error.

Figs. 95 and 96 show two instruments which work on a somewhat different principle. The first does not allow for the yielding of the supports, while the second works between collars on the specimen and is therefore independent of any movement of the supports. The instrument shown in Fig. 95 is also often used as a deflectometer in transverse testing.



CHAPTER V.

STRENGTH OF MATERIALS.—METHODS OF TESTING.

66. Standard Methods. —The importance of standard methods of testing material can hardly be over-estimated if it is desired to produce results directly comparable with those obtained by other experimenters, since it is found that the results obtained in testing the strength of materials are affected by the methods of testing and by the size and shape of the test-specimen. To secure uniform practice, standard methods for testing various materials have been adopted by several of the engineering societies, as well as by associations of the different manufacturers. The general and special standard methods adopted by these associations form the basis of methods described in this chapter.

67. Tension Testing. —*Form of Test-pieces.* —The form of test-pieces is found to have an important bearing on the strength, and for this reason engineers have adopted certain standard forms to be used. The form recommended by the Committee on Standard Tests and Methods of Testing, of the American Society of Mechanical Engineers, is as follows: *

"Specimens for scientific or standard tests are to be prepared with the greatest care and accuracy, and turned as nearly as possible according to the following dimensions. The tension test-pieces are to have different diameters according to the original thickness of the material, and to be, when expressed in English measures, exactly 0.4, 0.6, 0.8, and 1.0 inch in diameter; but for all these different diameters the angle, but not the length, of the neck is to remain constant. This neck is a cone, not a fillet connecting the shoulders and body. The length of the gauged or measured part to be 8 inches, of the cylindrical part 8.8 inches. The length of the coned neck to be 2½ times the diameter, increasing in diameter from the cylindrical part to 1¼ times the cylindrical part. The shoulders to have a length equal to the diameter, and to be connected with a round fillet to a head, which has a diameter equal to twice that of the cylinder, and a length at least 1¼ the diameter."

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* See Vol. XI. of Transactions.

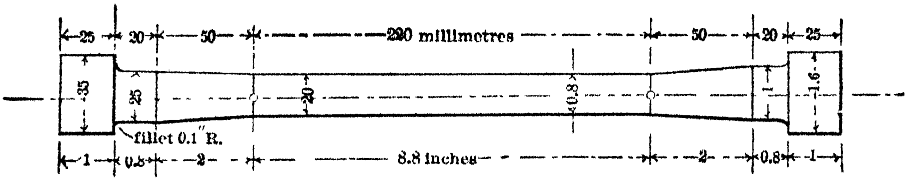

Fig. 97.—Standard Test-piece in Tension.

Fig. 97 shows the form of the test-piece recommended for tension; the numbers above the figure give dimensions in millimeters, those below in inches. For *flat test-pieces* the shape as shown in Fig. 98 is recommended: such specimens are to be cut from larger pieces; the fillets are to be accurately milled, and the shoulders made ample to receive and hold the full grip of the shackles or wedges.

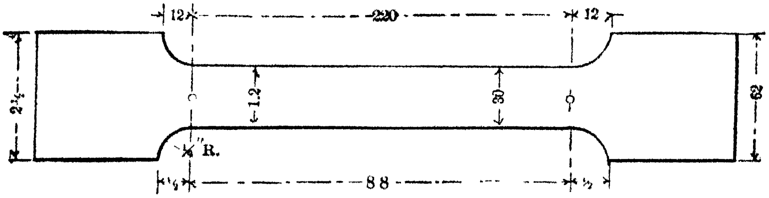

Fig. 98.—Test-piece for Flat Specimens.

If wedges are used for fastening, the ends to be gripped should be at least 2 inches long.

The length for rough bars when such are used as test-pieces is to be the same as for finished test-pieces, but the length of specimen from the gauge-mark to the nearest holder is to be not less than the diameter of the test-piece if round, or one and a half times the greatest side if flat.

For commercial testing the standard form cannot always be adhered to, and no form is recommended. The commonest form of test section in commercial work is ½ inch diameter and 2 inches gauge length. The ductilities found on this form of piece are much greater than would be given by the same material in the standard test-piece form. Ductility is always a function of the ratio of length to diameter of the test section. The strength determinations are nearly independent of the shape of the test section so long as the gauge-length exceeds twice the diameter; with a shorter gauge-length the strengths will be too high.

68. Test-pieces of Special Materials.—*Wood.*—Wood is a difficult material to test in tension, as the specimen is likely to be crushed by the shackles or holders. The author has had fairly good success with specimens, made with a very large bearing-surface in the shackles, of the form shown in Fig. 98, for flat specimens, but with the breadth of the shoulders or bearing-surfaces increased an amount equal to one-half the width of the specimen over that shown in Fig. 98.

Cast iron. — Cast-iron specimens of the usual or standard forms are very likely to be broken by oblique deformations in tension-tests much before the true tensile strength has been reached. To insure perfectly axial strains Riehlé Bros. propose a form of specimen

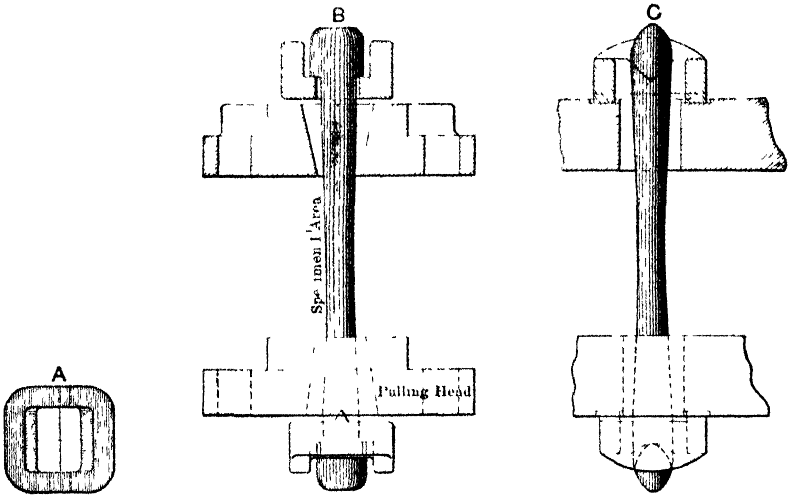

FIG. 99. — CAST-IRON SPECIMEN AND SELF-CENTERING GRIPS.

shown in Fig. 99, *A*, *B*, and *C*, cast with an enlarged head, the projecting portion of which, as shown in *C*, has a knife-edge shape. The specimen is carried in holders or shackles, *A* and *B*, which rest on knife-edges extending at right angles to those of the specimen.

This permits free play of the specimen in either direction, and renders oblique deformation nearly impossible.

Chain. — In the case of chain, large links are welded to the ends, as shown in Fig. 100; these are passed through the heads of the testing-machine and held by pins.

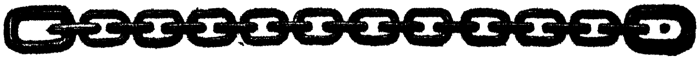

FIG. 100. — CHAIN TEST-PIECE.

Hemp Rope. — A similar method is used in testing hemp rope, the specimen being prepared as shown in Fig. 101. Special hollow conical shackles have also been used with success for holding the rope.

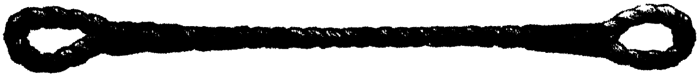

FIG. 101 — ROPE TEST-PIECE.

Wire Rope. — Wire-rope specimens may be prepared as shown in Fig. 102 or heads may be made by pouring a mass of melted babbitt metal around the ends of the wires arranged in an iron cone as in Fig. 103.

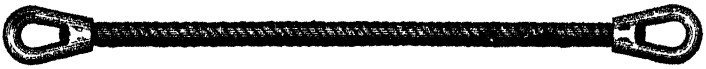

FIG. 102. — WIRE ROPE TEST-PIECE.

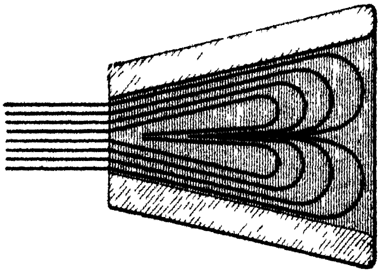

FIG. 103 — METHOD OF HOLDING ROPE.

69. Directions for Tension-tests.—Examine the test-piece carefully for any flaw, defect, irregularity, or abnormal appearance, and see that it is of correct form and carefully prepared. Indentations from a hammer often seriously affect the results. In wood specimens, abrasions, slight nicks at the corners, or bruises on the surface will invariably be the cause of failure.

Next, carefully determine the dimensions, record total length, gauge-length (or length on which measurements of strains are made), also form and dimensions of shoulders. Divide the specimen between the gauge-marks into inches and half inches, which may be marked with a special tool, or by rubbing chalk on the specimens and marking each division with a steel scratch. Special gauges, as shown in Figs. 104, 105, and 106, are convenient for this purpose. These marks serve as reference points in measuring the elongation after rupture, and this elongation should be measured, not from the center of the specimen, but either way from the point of rupture, as explained below.

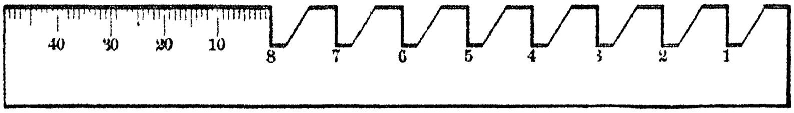

FIG. 104 MARKING GAUGE

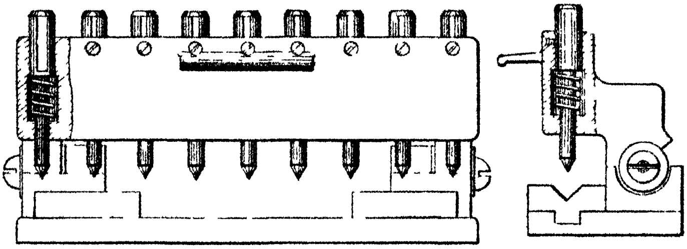

FIG. 105. — RIEHLÉ LAYING-OFF GAUGE

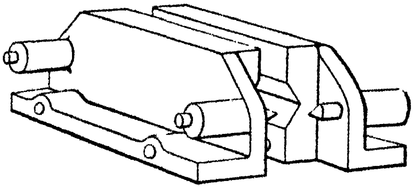

FIG. 106. — RIEHLÉ MARKING GAUGE

The matter of properly holding the specimen is of the greatest importance. Unless special forms of test-pieces or self-aligning grips are used, great care should be taken in the proper position of the wedges. Fig. 107, *a* to *e*, published by the Riehlé Co. shows the conditions that may exist when ordinary wedges are used. Fig. *a* shows the proper position, both wedges and specimen bearing the whole length. Fig. *b* shows what may occur with a thin specimen. This grip may hold if the load is not too high, but it is better to improve it by the use of liners, as in Fig. *c*. Fig. *d* shows a condition that should *never* be allowed to exist. The grip is almost sure to fail before the test is complete by the crushing of the top of the specimen, and above all the severe wedge action taking place may split the head of the machine.

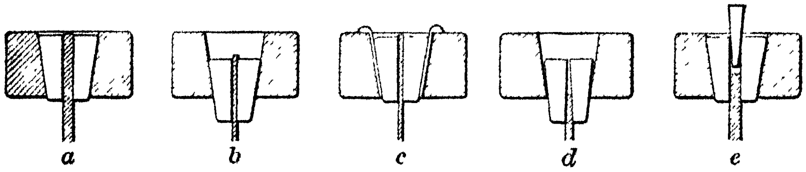

FIG. 107.—WEDGE ADJUSTMENTS.

A cure for the trouble is indicated in Fig. *e*, but the grip is not likely to stand high loads.

See that the testing-machine is level and balanced before each test; insert the specimen in a truly axial position in the machine.

Attach the auxiliary apparatus for measuring stretch, or obtaining autographic diagrams. The method of attaching extensometers will depend on the special form used, but this act should always be carefully performed, and the specimen exactly centered in the extensometer, and the gauge-points arranged 8 inches apart. The following directions for applying and using the Henning extensometer will serve to show the method to be used in all cases.

The Henning extensometer (see Fig. 83) is attached and used as follows: Before attaching the instrument, adjust the knife-edges in the clamps by means of the two milled nuts so that they are equally distant from the frame and a distance apart a little smaller than the diameter of the test-piece. Then, since the springs acting on the knife-edges are of equal strength, the instrument will adjust itself in the plane of the screws symmetrically with respect to the test-piece. Advance or withdraw the set-screws until their points are equally distant from the frame and far enough apart to admit the test-piece.

Separate the upper portion of the instrument, put it around the test-piece (already inserted in the machine), near the upper shoulder, with the smaller part to the right, force together and fasten securely. Advance the set-screws simultaneously until their points indent the test-piece. Separate the lower portion, put it around the test-piece with the vertical scales to the front, force together and secure. Hang the links on the proper bearings on both portions of the instrument. Then advance the set-screws as above. Throw the links out, take readings of the micrometers, apply the first increment of load, and proceed with the test as directed. To read the micrometers make the electrical connections; advance one micrometer until the bell rings announcing contact, back off barely enough to stop ringing. Read that side of the instrument. Back that side off further, and run up the other side and read as with the first. Read *break* of circuit. The vertical scale and the micrometer head are graduated so that readings to $\frac{1}{10000}$ inch can be obtained directly.

Calculate from the supposed coefficients of the material the probable load at elastic limit. Take one-tenth of this as the increment of load. The Committee on Standard Tests, American Society of Mechanical Engineers, recommend that the increment be one-half or one-third that of the probable load at the elastic limit, thus giving larger strains but fewer observations. Apply one increment of load to the specimen before measurements of elongation are made, since by loading specimens up to 1000 or 2000 pounds per square inch the effect of initial errors, such as occur generally at the commencement of each test, are lessened. The auxiliary apparatus adjusts itself somewhat during this period of loading, and the specimen assumes a true position should any slight irregularity exist. After passing the elastic limit, which will be indicated by the jump in stretch as shown by the extensometer, the test is run by applying the stress continuously and uniformly without intermission until the instant of rupture, only stopping at intervals long enough to make the desired observations of stretch and change of shape. The stress should at no time be decreased and reapplied in a standard test, but should be maintained continuously. The auxiliary apparatus for measuring strain must be removed before rupture takes place, except it is of a character not likely to be injured. It should usually be taken off very soon after the elastic limit is passed; although for ductile material it may be left in place for a longer time after the elastic limit has been passed than for hard and brittle materials. After the removal of the extensometer the stretched length of the test section may be measured with dividers. The material is to be loaded until fracture takes place, keeping the beam floating, after which the distortion for each part is to be measured by comparison with the reference divisions on the test-piece, measured from the point of rupture as previously explained. It is to be noted that measurements within the elastic limit are of especial importance, since materials in use are not to be strained beyond that point.

Remove the fractured piece from the machine; make measurements of shape, external and fractured surface; give time required in making the test.*

In recording the results of tests, loads at elastic limit, at yield-point, maximum, and instant of rupture are all to be noted.

The load at elastic limit is to be that stress which produces a change in the rate of stretch.

The load at yield-point is to be that stress under which the rate of stretch suddenly increases rapidly.

The maximum load is to be the highest load carried by the test-piece.

The load at instant of rupture is often not the maximum load, but a lesser load carried by the specimen at the instant of rupture.

In giving results of tests it is not necessary to give the load per unit section of reduced area, as such figures are of no value: (1) because it is not always possible to obtain the load at instant of rupture; (2) because it is generally impossible to obtain a correct measurement of the area of section after rupture; (3) lastly, because the amount of reduction of area may be dependent upon local and accidental conditions at the point of rupture. The modulus or coefficient of elasticity is to be deduced from measurements of deformation observed between fixed increments of load per unit section; as between

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* See Report of Committee on Standard Tests, Vol. XI., Am. Society Mech. Engrs

2000 pounds per square inch and 12,000 pounds per square inch; or between 1000 pounds per square inch and 11,000 pounds per square inch. With this precaution several sources of error are avoided, and it becomes possible to compare results on the same basis.

The character of the fracture often affords important information regarding the material. The structure of the fractured surface should be described as coarse or fine, either fibrous, granular, or crystalline. Its form, whether plane, convex, or concave, cup-shaped above or below, should in each case be stated. Its location should be accurately given, from marks on the specimen one-half inch or less apart. The reduction of diameter which accompanies fracture should be accurately measured. Accompanying the report should be a sketch of the fractured specimen.

Fracture occurs usually as the result of a gradual yielding of the particles of the specimen. The strain, so long as the stress is less than the maximum load, is distributed nearly uniformly over the specimen, but after that point is passed the distortion becomes nearly local; a rapid elongation with a corresponding reduction in section is manifest as affecting a small portion of the specimen only. This action in materials with sensible ductility takes place some little time before rupture; in very rigid materials it cannot be perceived at all. This peculiar change in form is spoken of as "necking."

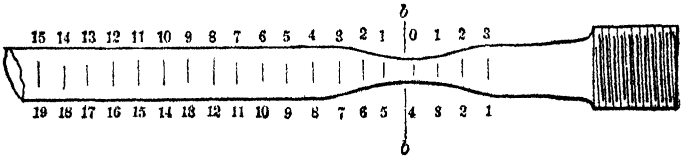

FIG. 108.

The drawing, Fig. 108, shows the appearance of a test-specimen in which the "necking" is well developed. Rupture occurs at *b*—*b*, a point in the neck which may be near one end of the specimen.

In order to measure the elongation of the specimen fairly, a correction should be applied, so that the reduced elongation shall be the same as though the stretch either side of the point of rupture were equal. This can only be done by dividing up the original specimen into equal spaces, each of which is marked so that it can be identified after rupture.

Suppose that twenty spaces represent the full length between gauge-marks: then if the rupture be at *b—b*, nearest the mark 0, Fig. 108, three spaces to the left of the nearest gauge-mark, the total length to compare with the original length should be 0 to 3 on the right, plus 0 to 10 on the left, plus the distance 3 to 10 on the left. These spaces are to be measured, and the sum taken as the total length after rupture. The stretch is the difference between this and the original length; the per cent of stretch, or elongation, is the stretch divided by the original length.

Report of Tension-tests. — In the report describe the testing-machine and method of testing, the form and dimensions of the specimen, the character and position of rupture. Make a sketch of the break. Submit a complete log of test (see blank form below). Calculate strength at elastic limit, maximum strength, and breaking strength, and moduli of elasticity and resilience. Plot a stress-deformation diagram on squared paper. Two scales for deformation should be used, one such that the elastic line shall have a slope of about 45° actual on the paper and the other such as to show the whole test. Both abscissa scalings should use the same scaling of ordinates.

The following form is used in Sibley College, Cornell University, for both tension- and compression-tests:

MECHANICAL LABORATORY—SIBLEY COLLEGE, CORNELL UNIVERSITY.

Test of..............................by......................................
Kind of Test...
Material from..
Machine used.............................Date.........................19
Time of Testing..........................min. Temp..............degrees F.

No.	Load.		Extensometer Readings.			Extension.			Modulus of Elasticity. E
	Actual. P	Per sq in. p	Right.	Left.	Mean.	Actual. λ	Difference. $\Delta \lambda$	Per in. ϵ	
1									
2									
18									
19									
20									

Original Length ....in., Diameter............in., Area...........sq. in
Final " .... " " .... " " .... "
Form of Section.............Fracture, Position........., Character............
Modulus of resilience..
Load per sq. inch, Elastic limit..............Max..........., Breaking..........
Equivalent elongation for 8 inches.......inches. Ductility..............per cent.
Reduction area.................per cent. Local elongation each half inch, from top,
1st.......2d.......3d.......4th.......5th.......6th... ...7th.......8th...
9th......10th......11th......12th......13th......14th ....15th.. ...16th.....

70. Compression Testing.—*Form of Test-pieces.*—Test-pieces are in all cases to be prepared with the greatest care, to make sure that the end surfaces are true parallel planes normal to the axis of the specimen.

1. *Short Specimens.*—The standard test-specimens for metals are to be cylinders 2 inches in length and 1 inch in diameter, when ultimate resistance alone is to be determined.

2. *Long Specimens.*—For all other purposes, especially when the elastic resistances are to be ascertained, specimens 1 inch in diameter and 10 or 20 inches long are to be used in the case of metal testing. Standard length on which deformation is to be measured is to be 8 inches, as in the tension-tests. Greatest care must be taken in all cases to insure square ends and that the force be applied axially.

The specimens are to be marked and the compression measured as explained for tension-test pieces.

71. Directions for Compression-tests.—1. *Short Pieces.*—In case of short pieces, measurements of deformation cannot be made on the test-piece itself, but must be made between points on the heads of the testing-machine. It is necessary to ascertain and make a correction for the error due to the yielding of the parts of the testing-machine. This is done as follows: Lower the moving-head until the steel compression-plate presses on the steel block on the lower platform with a force of about 500 pounds. Attach the micrometers to a special frame, which is supported by the upper platform, and read to a point on the movable head. With load at 500 pounds, read both micrometers. Apply loads by increments of 1000 pounds up to three-fourths the limit of the machine, taking corresponding readings. Plot a curve of loads and deflections with ordinates 1 long division = 1000 pounds, and abscissæ 1 long division = 0.001 inch. From this curve obtain corrections for the deflections caused by the loads used in the compression-test. In making the test calculate the increment of load as explained for tensile test, Article 69. Conduct the experiment in the same manner as for tension, except that the stress is applied to compress instead of to stretch the specimen. If the material tested is hard or brittle, as in cast iron, care should be taken to protect the person from the pieces which sometimes fly at rupture.

Report and draw curves as for tension-tests.

2. *Long Pieces.* — In this case the extensometers used for tension-tests can be connected directly to the specimen, and the measurements taken in substantially the same way, except that the heads of the extensometer will approach instead of recede from each other; this makes it necessary to *run the screws back each time after taking a measurement* a distance greater than the compression caused by the increment of load. In case large specimens are tested horizontally, initial flexion is to be avoided by counterweighting the mass of the test-piece.

Calculate the increment of load as one-tenth the breaking-load given by Rankine's formula, Article 43, page 56. Apply the first increment and take initial reading of micrometers; continue this until after the elastic limit has been passed, after which remove the extensometer, and apply load until rupture takes place. Protect yourself from injury by flying pieces. Compute the breaking coefficient C by Rankine's formula, and compare with the value usually assumed.

Compute the modulus of elasticity by $\frac{p}{\varepsilon}$ as in the tension-test.

Note in the report load at elastic limit, yield-point, and ultimate resistance, as well as increase of section at various points, and total compression, are calculated as explained for tension.

Submit a load-deformation diagram, and follow the same general directions as prescribed in the report for tension-testing.

72. **Transverse Testing.** — *Form of Test-pieces.* — For standard transverse tests, bars one inch square and forty inches long are to be used, the bearing blocks or supports to be exactly thirty-six inches apart, center to center. For standard or scientific tests of cast iron, such bars are to be cut out of a casting at least two inches square or two and a quarter inches in diameter, so as to remove all chilling effect. For routine tests, bars cast one inch square may be used, but all possible precautions must be taken to prevent surface-chilling and porosity.

Test-bars of wood are to be forty inches in length, and three inches square in section.

73. **Directions for Transverse Tests.**—Arrange a tension-compression machine for the transverse test by putting in supporting abutments and a loading head in the center, or use a special transverse-testing machine. The test-piece is usually a prismatic beam, loaded on a three-foot length, the load being applied at the center. The data required are loads and deflections at the center.

Sharp edges on all bearing-pieces are to be avoided, and the use of rolling bearings which move accurately with the angular deflections of the ends of the bars is recommended; otherwise the distance between fixed supports measured along the axis of the specimen is continually changing.

Place the test-bar upon the supports, and adjust the latter thirty-six inches apart between centers, and so that the load will be applied exactly at the middle. Obtain the necessary dimensions, and calculate the probable strength at elastic limit and at rupture by means of the formula $p = Ple \div 4I$. (See Article 44, page 60.) Adjust the specimen in the machine in a horizontal plane, and apply the stress at **the center,** normal to the axis of the specimen.

Measure the deflections at the center from a fixed plane or base, allowing for the settling of the supports, or by a special deflectometer. (See Article 65, page 111.)

Balance the scale-beam with the test-bar in position and the deflectometer lying on the platform. Set the poise for one increment of load and apply stress until the beam tips. Place the poise at zero, and balance by gradually removing the load. Place the deflectometer in position on the supports, and with the micrometer at zero make contact and record zero-reading and zero-load.

Apply the load in uniform increments equal to about one-tenth the calculated load for the elastic limit, stopping only long enough to measure the deflections. Wrought-iron is to be strained only until it has a sensible permanent set, but cast iron and wood are to be tested to rupture.

74. **Report of Transverse Tests.**—In the report describe the machine, method of making test, form of cross-section, peculiarities of the section, and make a sketch showing position and form of rupture.

Submit a complete log of the test together with a stress-deformation curve. Calculate the unit stresses and deformations by the formulas given in Article 44, table 1, page 60.

The following is a form for data and results of a transverse test:

MECHANICAL LABORATORY — SIBLEY COLLEGE, CORNELL UNIVERSITY.

Log and Report of Transverse-test.

Material.......................................from..............................
Breadth, *b*.....................in. Height, *h*,.....................in.
Length between supports, *l*,............in. Max. Fiber distance, *e*,..............in.
Moment of inertia, *I*,..............
Load applied at center of test length.
Testing Machine..........................
Time.............. hrs. .... min.
Date.............................. 19. ...
Observers { ....
....
....
....

No.	Load, *P*.	Stress per □″ in Outer Fiber *p*.	Deflection.		Relative Deformation in Outer Fibers.		Modulus of Elasticity, lbs. per sq. in., *E*.
			Reading, inches.	Total Deflection, in., *d*.	Calculated *ε′*.	Corrected for Zero Error, *ε*.	
1							
2							
3							
4							
5							
6							
7							
8							
9							
10							
11							
12							
13							
14							
15							
16							
17							
18							
19							
20							

	Actual Load, lbs.	Deflection, inches.	Stress per sq. in. in Outer Fiber.	Relative Deformation in Outer Fiber.	Greatest Vertical or Horizontal Shear Stress, lbs. per sq. in.
At Elastic Limit					
At Maximum Load					

Modulus of Elasticity, E,..............lbs. per sq. in. Modulus of Resilience, U,.............in. lbs. per cu. in.

Formulas (for central loading):

These formulas are true only for elastic action of the material.

$$\begin{cases} p = \frac{c}{I} \cdot M = \left(\frac{6}{bh^2}\right) \cdot M = \frac{3}{2} \cdot \frac{l}{bh^2} \cdot P \\ \epsilon' = \frac{12c}{l^2} \cdot d = \frac{6h}{l^2} \cdot d \\ \epsilon = \epsilon' + \text{zero correction.} \\ E = \frac{\Delta p}{\Delta \epsilon} \cdot \quad U = \tfrac{1}{2} \cdot p \cdot \epsilon \text{ at elastic limit (at break for brittle materials).} \end{cases}$$

75. **Torsion Testing.** — *Form of Test-pieces.* — For standard tests, cylindrical specimens with cylindrical concentric shoulders are to be used; the two are connected by large fillets. The specimen is to be held in the heads of the machine by three keys, inserted in key-ways $\frac{1}{8}$ inch deep, cut in the shoulder. In commercial testing the ends of the specimen are usually held in self-centering jaw-chucks on the heads of the testing-machine.

76. **Directions for Torsion-Tests.** — 1. *With Olsen or Riehlé Machines.* — The general method of test is like that of the transverse test, as to choice of increment of load, etc. For measuring elastic deformations arms are attached to the piece as described in Article 59, page 93. For measuring the deformation at the break there should be scribed on the piece before loading a line parallel to the axis. After the break the distance along the piece in which this line makes one or more complete turns can easily be measured, and by proportion the number of turns, or angle of twist, in the test-length can then be computed. The final dimensions of the test-section should be taken.

The report should describe the test-piece and testing-machine, method of test, and action of the material under stress. Note position and character of fracture. Calculate the unit stress and deformation in the outer fiber. Submit a log of test (see form below) and a curve showing the variation of unit stress against unit deformation, similar to the tension-test. Calculate the moduli of elasticity and resilience in shear. The following is a form for reporting a torsion-test:

MECHANICAL LABORATORY — SIBLEY COLLEGE, CORNELL UNIVERSITY.

Log and Report of Torsion-test.

Material.................................From ....
Diameter, d,............in. Max. Fiber distance, r, ...in.
Length between measuring arms, l. ......in. Length of measuring arms, L,.........in.
Testing machine.........................
Time...........hrs. ....min. Observers ....
....
....
....
Date .... 19. ..

No.	Moment of Torsion, in.-lbs. = M	Shearing Stress, per □ inch at surface = q	Angle of Torsion. Scale Reading, inches.	Angle of Torsion. Distance on Arc, inches = A	Angle of Torsion. Total π Measure = a	Helix Angle at Surface, π Measure = δ	Modulus of Rigidity, pounds per sq. in. = E_s
1							
2							
3							
4							
5							
6							
7							
8							
9							
10							
11							
12							
13							
14							
15							
16							
17							
18							
19							
20							

	Moment of Torsion	Angle of Torsion	Shearing Stress, lbs per sq. in at Surface.	Helix Angle at Surface.
Elastic Limit .	$M_E =$	$a_E =$	$q_E =$	δ_E..........
Maximum	$M_m =$	$a_m =$	$q_m =$	δ_m ....

Modulus of Rigidity, E_s, average,.... ..lbs per sq. in. Modulus of Resilience, U, ...in. lbs per cu. in.

Formulas

Within or near elastic limit

$$q = \frac{2}{\pi r^3} \cdot M = \frac{16}{\pi d^3} \cdot M$$

$$\delta = \frac{r}{l} \cdot a \qquad a = \frac{A}{L}$$

$$E_s = \frac{\Delta q}{\Delta \delta}$$

$$U = \tfrac{1}{2} q_E \delta_E$$

At maximum load (break)

q_m = modulus of rupture = $\frac{2}{\pi r^3} \cdot M_m$

q_s = true shear stress in surface, for ductile materials, = $(q_m - \tfrac{1}{3} q_E)$

$$\delta_m = \tan^{-1} \frac{r}{l} \cdot a$$

2. *With Thurston Autographic Machine.* — Special test-pieces are used, generally with a test section about ⅝ inch diameter and 2 inches or less long, with the ends of the piece square to hold in the wedges. The pieces are generally turned from square stock of size somewhat greater than the desired diameter of test section.

Determine first the maximum moment of the pendulum. This may be done by swinging the pendulum so that its center-line is horizontal, supporting it on platform-scales and taking the weight and the distance of the point of support from the center of suspension of the pendulum. The product of these two quantities is the maximum moment of the pendulum. Make three determinations, using different lever-arms, and take the mean for the true moment of the pendulum. A correction for the friction of the journal of the pendulum must be made. When hanging vertically, measure with a spring-balance, inserted in the eye near the bob, the force necessary to start the pendulum. Add this moment to that obtained above, and the result is the total maximum moment of the pendulum. From this the value of the moment for any angular position may be calculated.

Note the variation of position of the pencil-point between the vertical and the horizontal positions of the pendulum. This distance laid down on the Y-axis of the record-sheet corresponds to the maximum moment obtained above, whence calculate the value of one inch of ordinate. Calculate the length corresponding to one degree on the surface of the paper drum, parallel to the X-axis. This will be the unit to be used in calculating the angle of torsion. Fix the paper on the drum and draw the datum-line or X-axis. Insert the test-piece between the centers and screw in the center until the neck of the test-piece is about midway between the jaws. Wedge the test-piece between the jaws as firmly as possible by hand, and then tap the wedges slightly with a copper hammer. Throw the worm into gear and turn the handle slowly and steadily until rupture occurs, unless set-lines are taken. Take the record of all the test-pieces on the same sheet with the same origin of co-ordinates. The diagram is drawn by attachment to the working parts of the frame, and consequently any yielding of the frame or slipping of the jaws appears on the diagram as a strain or yield of the specimen. The angular deformation α, as obtained from the diagram, is likely to be too great, especially within the elastic limit.

The characteristic form of diagram given by the torsion-machine is shown in Fig. 109, in which the results of tests of several materials are shown. In these diagrams the ordinates are moments of torsion (M), the abscissæ are developments of the angle of torsion (α).

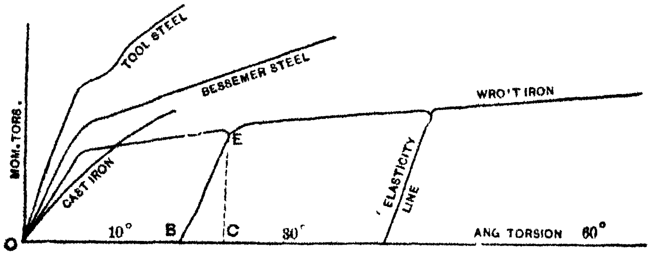

FIG. 109

Save that measurements from these autographic diagrams take the place of data and curves of the usual testing, the method of working up the test is as given above.

77. **Testing in Direct Shear.** — Satisfactory direct-shear tests are difficult to make. In the commonest form of this test, a pin of material is broken in double shear, by fitting it through three links, pulling or pushing the outer links past the central one. The arrangement must be rigid and must not allow either shearing force to acquire a lever-arm; yet friction between the links themselves must be small. More information about the properties of a material under shear stressing can be had from a torsion-test than from a test in direct shear. The latter test gives the ultimate strength only; the torsion-test gives the elastic strengths and moduli.

78. **Impact Testing.** — No universally acceptable impact-testing machine has yet been devised. The results of impact tests vary from machine to machine, as well as from material to material.

The following are directions for testing cast-iron with Heisler's impact-testing machine. (See Article 60, p. 95.) Take a transverse test-bar of cast-iron and place it in the machine, cope side out, so that the blow will be struck in the middle of its length. Arrange the autographic device so that it will register the deflection of the bar.

Place the tripping device or "dog" for a fall of two inches. Catch the bob at this point, and trip at every notch above successively until the bar breaks. Note the maximum height of fall. Report on the experiment the behavior of the test-bar and character of its fracture, and the number of impacts and the force in inch-pounds of the last blow. Compute the resilience of the test-piece. Try a similar bar at same ultimate fall, and observe the number of blows required to break it. Draw conclusions. Write complete report, and give moduli and coefficients.

79. **Drop-tests.** — Drop-tests are a form of impact test used on railway materials, as car wheels, axles, etc. The specification calls for the taking without rupture of a certain number of blows from a certain height. The following method of making drop-tests has been recommended by the Committee on Standard Methods of Testing appointed by the American Society of Mechanical Engineers, and is substantially the same as adopted by the German Engineers at Munich in 1888:

Drop-tests are to be made on a *standard drop*, which is to embody the following essential points:

a. Each drop-test apparatus must be standardized.

b. The "ball" (*falling mass*) shall weigh 1000 or 1500 pounds; the smaller is, however, preferable.

c. The "ball" may be made of cast iron, cast or wrought steel; the shape is to be such that its center of gravity be as low as possible.

d. The striking-block is to be made of forged steel, and is to be secured to the "ball" by dovetail and wedges in a rigid manner, and so that the striking-face is placed strictly symmetrical about and normal to its vertical axis passing through the center of gravity. Special permanent marks are to indicate the correctness of the face in these respects.

Special marks should be made to indicate the center of the anvil-block.

e. The length of guides on the ball should be more than twice the width between the guides, which are to be made of metal; i.e., rails so placed that the ball has but a minimum amount of play between them. Graphite is recommended as lubricant.

f. The detachment or shears must not cause the ball to oscillate between the guides, and must be readily and freely controllable, with the point of suspension truly above the center of gravity of the ball; and a short movable link, chain, or rope is to be fixed between the ball and shears or detachment.

g. When a constant height of drop is used, an automatic detaching device is recommended.

h. The bearings for the test-piece are to be rigidly attached to the scaffold or frame, and they should be, wherever possible, in one piece with it.

i. The weight of frame, bearings, and anvil-block should be at least ten times that of the ball.

j. The foundation should be inelastic, and consist of masonry, the dimensions of which are to be determined by the locality and subsoil.

k. The surface struck should always be accurately level; therefore proper shoes or bearing-blocks are to be provided for testing rails, axles, tires, springs, etc., etc., to insure a proper level upper surface; these blocks are to be as light as possible.

The exact shape of these bearing-blocks is to be given in each test report.

l. The gallows or frame should be truly vertical and the guides accurately parallel.

m. The height of fall of ball should be 20 feet clear, between striking and struck surfaces.

n. Drops which by friction of ball on guides absorb two per cent of the work due to impact are to be discarded.

o. For tests on large specimens a ball weighing 2000 pounds is to be used.

p. A sliding-scale is to be attached to the frame, and in such a manner that the zero-mark can always be placed on a level with the top of the test-piece.

80. Minor Tests: *The Welding-test.* — The welding is to be done with a hammer weighing eight to ten pounds, with a given number of blows. The weld is to be a simple scarf weld, made in a coke or gas flame without fluxes. Each bar to be tested is to be treated in the same way, using in each case two or three samples of iron; one weld is to be tested on the tension-machine, the other to be nicked to the depth of the weld and then bent or broken, to show the character of the welded surfaces.

The Bending-test. — This affords a ready means of finding the ductility of metals. The test-piece is to be bent about a stud having a diameter twice that of the specimen. The piece is to be bent with a lever, and no pounding is permitted. If the plate holding the stud is graduated, the angular deflection at time of permanent set may be read at once. A modification of the bending-test is often used to determine the property of toughness, by bending the specimen, first hot and then cold, until it is doubled over on itself.

The Drifting-test. — This tests the softness and ductility of metals. It is made by driving a "drift," a tapered steel punch, so as to enlarge a hole drilled in a plate. The measurement is in the amount of enlargement of the hole before the edges crack and tear.

The Punching-test. — Find the least material that will stand between the edge of the plate and the hole punched, by measurement.

The Forging-test. — The material is brought to a red heat and hammered until cracks begin to show, the relative amount of flattening indicating the red-shortness of the material. Useful principally with rivet-rods.

The Hammer-test. — This is made with a light hammer, and the character of the material is determined by the sound emitted. Is useful in locating defects in finished products, but of little value on test-specimens.

The Hardening-test is made by heating a specimen to cherry-red heat and plunging it in water having a temperature of 32 to 40 degrees. The specimen so treated is tested by bending or torsion, the same as an unhardened specimen. Used for boiler plate and similar materials to find whether the carbon content is too high.

The Abrasion-test. — Find the amount of wear from a given amount of work. For testing paving-brick and similar material special abrasion-testing machines have been devised, and the test is standardized. These machines are of barrel form. A charge is put in consisting of a certain number of brick and of hard cast-iron cubes of 1-inch or 2-inch face. The charged barrel is rotated a certain number of times at a certain speed, such that its contents get a thorough mixing and shaking. The measurement is in percentage of weight lost by the brick.

Abrasion-tests of metals, generally bearing metals, are made by holding a block of the metal against a hardened steel wheel of large diameter, using a certain pressure between the block and the wheel, running the wheel a definite number of revolutions, and measuring the loss of weight or thickness of the block tested.

The Hardness-test. —The "hardness" of a material is usually understood as a relative property, having reference to the ability of one material to cut another. Hardness is a composite property, depending partly on density and on tensile and shear strengths. Hence the machines devised to measure hardness are also strength-measuring devices. Two types of hardness-testers are likely to be met with by the engineer, the pressure and the impact types, represented respectively by the Brinell ball tester and the Shore "Scleroscope."

1. *The Brinell Ball-testers.* —For the Brinell ball-tester a flat of an inch or so in diameter is prepared on the surface of the material to be tested. Then a hard steel ball, usually five-eighths of an inch in diameter, is pressed into the material with a certain load. The depth or diameter of the impression made in the material is then measured with a special micrometer. From this, the load used, and the size of the ball used, the "hardness" is computed. These same data have been found to be so intimately connected with the ultimate strength of the material that this quantity also can be computed with good accuracy from the ball-test. The close connection with the ultimate strength can be understood when one considers that to displace the material and make the

2. *The Scleroscope.* — The Shore "Scleroscope" drops onto a small flat surface of the material to be tested a tiny diamond-pointed steel hammer. If the material were perfectly elastic and no work were done on it (perfect hardness), the hammer would rebound to the height from which it fell. If some of the energy of the hammer is used up in doing work on the material tested, making an impression in the material, then the rebound of the hammer is not complete. The softer the material, the less is the rebound. Hence the ratio of height of fall to height of rebound of the hammer measures the hardness of the material. On account of the rapidity of the blow and the smallness of the area of material affected, the strength with which the material resists deformation by the scleroscope hammer is the elastic strength of the material.

On brittle materials, where the elastic strengths and ultimate strengths are close together, the two types of hardness-testers will give parallel results. On ductile materials the ball-tester seems to get nearer to the cutting hardness. The impact type of testers will show a "hardness" more than proportionate to the cutting hardness on cold-worked metals, on account of the elevation of the elastic strength by cold working.

The Fatigue-test. — Fatigue-tests are made to determine the resistance of materials to repeatedly applied loads, either with or without reversal of stress. No commercially satisfactory fatigue-test has yet been devised. It is well established that failure will occur under repeated loads of much less intensity than is needed for failure under single loading. This is due to (1) the time factor in shear failure, so that the material tested in an ordinary "single-loading" tensile test does not break down at the yield-point at as low a stress as it should; (2) the nonhomogeneity of nature of the metals, being different in properties in adjacent crystals of their make-up, and the repeated loadings searching out the weak spots

81. Special Tests and Specifications. — In general the material is to be tested in such a manner as to develop the same properties that will be called forth in the peculiar use to which it is devoted.

The table below shows the tests that are prescribed for materials for various uses, by the Committee on Standard Tests and Methods of Testing of the American Society of Mechanical Engineers:

TABLE SHOWING TESTS REQUIRED

Required Test Denoted by x

Material used for	Tension	Compression	Transverse	Torsion	Impact	Welding	Bending	Hardening	Forging	Abrasion	Punching
Railroad rails	x				x		x				
" car-axles	x				x		x				
" " tires	x				x						
Shafting	x		x	x							
Building — wrought iron	x	x	x			x	x				
" low steel	x	x	x				x	x			
" high steel	x	x	x				x	x			
Boiler — wrought iron											
" plates	x		x				x		x		x
" shape iron	x					x	x		x		x
" rivet-rods	x						x		x		
" low steels	x						x	x	x		
Ship materials											
" plates	x						x				
" rivets	x						x		x		
Wire	x						x*				
Wire rope	x				x†						
Cast iron	x	x					x				
Copper and soft metals	x	x					x				
Woods	x	x	x								
Stones		x	x								

* Repeat in both directions — also by winding † Longitudinal

Many of the Engineering Societies and of the different associations of manufacturers have formulated and adopted specifications for, and methods of test of, the different engineering materials. These rules differ more or less among themselves even in the case of materials intended for the same purposes. It is therefore necessary to determine, before testing, just what set of specifications a material is to meet. Some of the more important sets of rules are given below, but, as these rules are modified from time to time, those cited must be considered rather as examples of what to expect than as specifications to be used in any individual case without further investigation.

MANUFACTURERS' STANDARD SPECIFICATIONS.

(Revised to Feb 6, 1903.)

1. STRUCTURAL STEEL.

Process of Manufacture.

1. Steel may be made by either the Open-hearth or the Bessemer process.

Testing and Inspection

2. All tests and inspection shall be made at the places of manufacture prior to shipment.

Test-pieces.

3 The tensile strength, limit of elasticity and ductility, shall be determined from a standard test piece cut from the finished material The standard shape of the test-piece for sheared plates shall be as shown by the following sketch (Fig. 110).

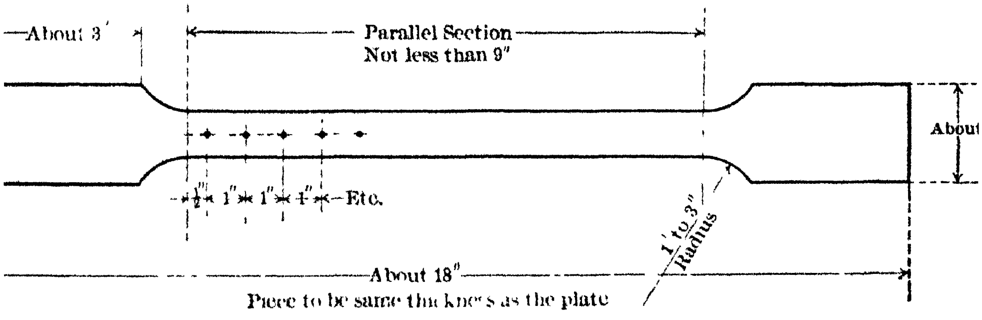

FIG. 110.

On specimens cut from other material the test-piece may be either the same as for sheared plates, or it may be planed or turned parallel throughout its entire length, and in all cases where possible, two opposite sides of the test-piece shall be the rolled surfaces. The elongation shall be measured on an original length of 8 inches except as modified in section 12, paragraph c Rivet rounds and small bars shall be tested of full size as rolled

Two test-pieces shall be taken from each melt or blow of finished material, one for tension and one for bending; but in case either test develops flaws, or the tensile test-piece breaks outside of the middle third of its gauged length, it may be discarded and another test-piece substituted therefor.

Annealed Test-pieces.

4. Material which is to be used without annealing or further treatment shall be tested in the condition in which it comes from the rolls. When material is to be annealed or otherwise treated before use, the specimen representing such material shall be similarly treated before testing.

Marking.

5. Every finished piece of steel shall be stamped with the blow or melt number, and steel for pins shall have the blow or melt number stamped on the ends. Rivet and lacing steel, and small pieces for pin plates and stiffeners, may be shipped in bundles securely wired together, with the blow or melt number on a metal tag attached.

Finish.

6. Finished bars shall be free from injurious seams, flaws or cracks, and have a workmanlike finish.

Chemical Properties.

7a. Steel for buildings, train sheds, highway bridges and similar structures. } Maximum phosphorus, 0.10 per cent.

7b. Steel for railway bridges: Maximum phosphorus, 0.08 per cent.

Physical Properties.

8. Structural steel shall be of three grades: *Rivet, Railway Bridge* and *Medium.*

Rivet Steel.

9. Ultimate strength, 48,000 pounds per square inch. Elastic limit, not less than one-half the ultimate strength.

Percentage of elongation, $\frac{1,400,000}{\text{Ultimate strength}}$.

Bending-test, 180 degrees flat on itself, without fracture on outside of bent portion.

Steel for Railway Bridges.

10. Ultimate strength, 55,000 to 65,000 pounds per square inch. Elastic limit, not less than one-half the ultimate strength.

Percentage of elongation, $\frac{1,400,000}{\text{Ultimate strength}}$.

Bending-test, 180 degrees to a diameter equal to thickness of piece tested without fracture on outside of bent portion.

Medium Steel.

11. Ultimate strength, 60,000 to 70,000 pounds per square inch. Elastic limit. not less than one-half the ultimate strength.

Percentage of elongation, $\frac{1,400,000}{\text{Ultimate strength}}$.

Modifications in Elongation for Thin and Thick Material.

12. For material less than 5/16 inch and more than ¾ inch in thickness, the following modifications shall be made in the requirements for elongation:

a. For each increase of ⅛ inch in thickness above ¾ inch, a deduction of 1 per cent shall be made from the specified elongation, except that the minimum elongation shall be 20 per cent for eye-bar material and 18 per cent for other structural material.

b. For each decrease of 1/16 inch in thickness below 5/16 inch, a deduction of 2½ per cent shall be made from the specified elongation.

c. In rounds of ⅝ inch or less in diameter, the elongation shall be measured in a length equal to eight times the diameter of section tested.

d. For pins made from any of the before-mentioned grades of steel, the required elongation shall be 5 per cent less than that specified for each grade, as determined on a test-piece, the center of which shall be one inch from the surface of the bar.

Variation in Weight.

13. The variation in cross-section or weight of more than 2½ per cent from that specified will be sufficient cause for rejection, except in the case of sheared plates, which will be covered by the following permissible variations:

a. Plates 12½ pounds per square foot or heavier, up to 100 inches wide, when ordered to weight, shall not average more than 2½ per cent variation above or 2½ per cent below the theoretical weight. When 100 inches wide and over, 5 per cent above or 5 per cent below the theoretical weight.

b. Plates under 12½ pounds per square foot, when ordered to weight, shall not average a greater variation than the following:

Up to 75 inches wide, 2½ per cent above or 2½ per cent below the theoretical weight. Seventy-five inches wide up to 100 inches wide, 5 per cent above or 3 per cent below the theoretical weight. When 100 inches wide and over, 10 per cent above or 3 per cent below the theoretical weight.

2. STRUCTURAL CAST IRON.

1. Except when chilled iron is specified, all castings shall be tough gray iron, free from injurious cold-shuts or blow-holes, true to pattern, and of a workmanlike finish. Sample pieces, one inch square, cast from the same heat of metal in sand moulds, shall be capable of sustaining on a clear span of 4 feet 8 inches a central load of 500 pounds when tested in the rough bar.

3. SPECIAL OPEN-HEARTH PLATE AND RIVET STEEL.

Testing and Inspection.

1. All tests and inspections shall be made at the place of manufacture prior to shipment.

Test-pieces.

2. The tensile strength, limit of elasticity and ductility, shall be determined from a standard test-piece cut from the finished material. The standard shape

of the test-piece for sheared plates shall be as shown by the following sketch (see Fig. 111):

On specimens cut from other material the test-piece may be either the same as for sheared plates, or it may be planed or turned parallel throughout its entire length, and, in all cases where possible, two opposite sides of the test-piece shall be rolled surfaces. The elongation shall be measured on an original length of 8 inches, except as modified in section 12, paragraph *c*. Rivet rounds and small bars shall be tested of full size as rolled

Four test-pieces shall be taken from each melt of finished material, two for tension and two for bending; but in case either test develops flaws, or the tensile test-piece breaks outside of the middle third of its gauged length, it may be discarded and another test-piece substituted therefor.

Annealed Test-pieces.

3. Material which is to be used without annealing or further treatment shall be tested in the condition in which it comes from the rolls When material is to be annealed or otherwise treated before use, the specimen representing such material shall be similarly treated before testing.

Marking.

4. Every finished piece of steel shall be stamped with the melt number. Rivet steel may be shipped in bundles securely wired together, with the melt number on a metal tag attached

Finish.

5. All plates shall be free from injurious surface defects and have a workmanlike finish.

Chemical Properties.

6*a*. Flange or boiler steel.	Maximum phosphorus, 0.06 per cent. Maximum sulphur, 0.04 per cent.
6*b*. Extra-soft and fire-box steel.	Maximum phosphorus, 0 04 per cent. Maximum sulphur, 0 04 per cent.

Physical Properties.

7. Special Open-hearth Plate and Rivet Steel shall be of three grades: *Extra-soft, Fire-box* and *Flange* or *Boiler Steel*

Extra-soft Steel.

8. Ultimate strength, 45,000 to 55,000 pounds per square inch. Elastic limit, not less than one-half the ultimate strength. Elongation, 28 per cent. Cold and quench bends, 180 degrees flat on itself, without fracture on outside of bent portion.

Fire-box Steel.

9. Ultimate strength, 52,000 to 62,000 pounds per square inch. Elastic limit, not less than one-half the ultimate strength. Elongation, 26 per cent. Cold and quench bends, 180 degrees flat on itself, without fracture on outside of bent portion.

Flange or Boiler Steel.

10. Ultimate strength, 55,000 to 65,000 pounds per square inch.

Elastic limit, not less than one-half the ultimate strength.

Elongation, 25 per cent.

Cold and quench bends, 180 degrees flat on itself, without fracture on outside of bent portion.

Boiler-rivet Steel.

11. Steel for boiler rivets shall be made of the extra-soft grade specified in paragraph No. 8.

Modifications in Elongation for Thin and Thick Material.

12. For material less than $\frac{5}{16}$ inch, and more than ¾ inch in thickness, the following modifications shall be made in the requirements for elongation:

a. For each increase of ⅛ inch in thickness above ¾ inch, a deduction of 1 per cent shall be made from the specified elongation.

b For each decrease of $\frac{1}{16}$ inch in thickness below $\frac{5}{16}$ inch, a deduction of 2½ per cent shall be made from the specified elongation.

c. In rounds of ¾ inch or less in diameter, the elongation shall be measured in a length equal to eight times the diameter of section tested.

Variation in Weight.

13. The variation in cross-section or weight of more than 2½ per cent from that specified will be sufficient cause for rejection, except in the case of sheared plates, which will be covered by the following permissible variations:

a. Plates 12½ pounds per square foot or heavier, up to 100 inches wide, when ordered to weight, shall not average more than 2½ per cent variation above or 2½ per cent below the theoretical weight When 100 inches wide and over, 5 per cent above or 5 per cent below the theoretical weight.

b. Plates under 12½ pounds per square foot, when ordered to weight, shall not average a greater variation than the following:

Up to 75 inches wide, 2½ per cent above or 2½ per cent below the theoretical weight. Seventy-five inches wide up to 100 inches wide, 5 per cent above or 3 per cent below the theoretical weight. When 100 inches wide and over, 10 per cent above or 3 per cent below the theoretical weight.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION SPECIFICATIONS (Adopted 1898).

1. *Cast Iron.* — Should be of soft, gray texture and high degree of ductility. To be used only for hand-hole plates, crabs, yokes, etc., and manheads. It is a dangerous metal to be used in mud drums, legs, necks, headers, manhole rings, or any part of a boiler subject to tensile strains; its use is prohibited for such parts.

2. *Steel.* — Homogeneous steel made by the open-hearth or crucible processes, and having the following qualities, is to be used in all boilers.

Tensile Strength, Elongation, Chemical Tests. — Shell plates *not* exposed to the direct heat of the fire or gases of combustion, as in the external shells of internally fired boilers, may have from 65,000 to 70,000 pounds tensile strength; elongation not less than 24 per cent in 8 inches; phosphorus not over 0.035 per cent; sulphur not over 0.035 per cent.

Shell plates in any way exposed to the direct heat of the fire or the gases of combustion, as in the external shells or heads of externally fired boilers, or plates on which any flanging is to be done, to have from 60,000 to 65,000 pounds tensile strength; elongation not less than 27 per cent in 8 inches; phosphorus not over 0.03 per cent; sulphur not over 0.025 per cent.

Fire-box plates, or such as are exposed to the direct heat of the fire, or flanged on the greater portion of their periphery, to have 55,000 to 62,000 pounds tensile strength; elongation, 30 per cent in 8 inches; phosphorus not over 0.03 per cent; sulphur not over 0.025 per cent.

For all plates the elastic limit to be at least one-half the ultimate strength; percentage of manganese and carbon left to the judgment of the steel maker.

Test Section to be 8 inches long, planed or milled edges; its cross-sectional area not less than one-half of one square inch, nor width less than the thickness of the plate.

Bending-test. — Steel up to ½ inch thickness must stand bending double and being hammered down on itself; above that thickness it must bend round a mandrel of diameter one and one-half times the thickness of plate down to 180 degrees. All without showing signs of distress.

Bending test-piece to be in length not less than sixteen times thickness of plate and rough, shear edges milled or filed off. Such pieces to be cut both lengthwise and crosswise of the plate.

All tests to be made at the steel mill. Three pulling-tests and three bending-tests to be made from each heat. If one fails the manufacturer may furnish and test a fourth piece, but if two fail the entire heat to be rejected.

Certified copies of tests to be furnished each member of A. B. M. A. from heats from which his plates are made.

3. *Rivets.* — All rivets to be of good charcoal iron, or of a soft, mild steel, having the same physical and chemical properties as the fire-box plates, and must test hot and cold by driving down on an anvil with the head in a die; by nicking and bending, by bending back on themselves cold, without developing cracks or flaws.

4. *Boiler Tubes,* of charcoal iron or mild steel specially made for the purpose and lap-welded or drawn; they should be round, straight, free from scales, blisters, and mechanical defects, each tested to 500 pounds internal hydrostatic pressure.

This fact and manufacturer's name to be plainly stenciled on each tube.

Standard Thicknesses by Birmingham wire gauge to be
No. 13 for tubes 1 in., 1¼ in., 1½ in. and 1¾ in. diameter.
No. 12 for tubes 2 in., 2¼ in. and 2½ in. diameter.
No. 11 for tubes 2¾ in., 3 in., 3¼ in. and 3½ in. diameter.
No. 10 for tubes 3¾ in. and 4 in. diameter.
No. 9 for tubes 4½ in. and 5 in. diameter.

Tests. — A section cut from one tube taken at random from a lot of 150 or less must stand hammering down cold vertically without cracking or splitting when down solid.

Length of test-pieces:

¾ in. for tubes from 1 in. to 1¾ in. diameter.
1 in. for tubes from 2 in. to 2½ in. diameter.
1¼ in. for tubes from 2¾ in. to 3¼ in. diameter.
1½ in. for tubes from 3½ in. to 4 in. diameter.
1¾ in. for tubes from 4½ in. to 5 in. diameter.

All tubes must stand expanding flange over on tube plate and bending without flaw, crack, or opening of the weld.

5. *Stay Bolts* to be made of iron or mild steel specially manufactured for the purpose, and must show on:

Test Section, 8 inches long, net:

For Iron, tensile strength not less than 46,000 lbs.; elastic limit not less than 26,000 lbs.; elongation not less than 22 per cent for bolts of less than one (1) square inch area, nor less than 20 per cent for bolts one (1) square inch and more in net area.

For Steel, tensile strength not less than 55,000 lbs.; elastic limit not less than 33,000 lbs.; elongation not less than 25 per cent for bolts of less than one (1) square inch area, nor less than 22 per cent for bolts one (1) square inch and more in net area.

Tests. — A bar taken from a lot of 1000 lbs. or less at random, threaded with a sharp die "V" thread with rounded edges, must bend cold 180 degrees around a bar of same diameter, without showing any crack or flaws.

Another piece, similarly chosen, and threaded, to be screwed into well-fitting nuts formed of pieces of the plates to be stayed, and riveted over so as to form an exact counterpart of the bolt in the finished structure; to be pulled in testing-machine and breaking stress noted; if it fails by pulling apart the tensile stress per square inch of net section is its measure of strength; if it fails by shearing the shear stress per square inch of mean section in shear is this measure. The mean section in shear is the product of half the thickness of the plate by the circumference at half height of thread.

6. *Braces and Stays.* — Material to be fully equal to stay-bolt stock, and tensile strength to be determined by testing a bar not less than ten inches (10 in.) long from each lot of 1000 lbs. or less.

LLOYD'S SPECIFICATIONS FOR QUALITY AND TESTING OF SHIP STEEL (1907–1908).

1. *Process of Manufacture.* — Steel for shipbuilding shall be made by the open-hearth process, acid or basic.

2. *Freedom from Defects.* — The finished material shall be free from cracks, surface flaws and lamination. It shall also have a workmanlike finish, and must not have been hammer-dressed.

3. *Testing and Inspection.* — The following tests and inspections shall be made at the place of manufacture prior to dispatch; but, in the event of any of the material proving unsatisfactory in the course of being worked into vessels, such material shall be rejected, notwithstanding any previous certificate of satisfactory testing, and such further tests of the material from the same charge may be made as the Surveyor may consider desirable.

4. *Tensile Test-pieces.* — The tensile strength and ductility shall be determined from standard test-pieces cut lengthwise or crosswise from the rolled material. When material is annealed or otherwise treated before dispatch, the test-pieces shall be similarly and simultaneously treated with the material before testing.

Plates. — Wherever practicable the rolled surfaces shall be retained on two opposite sides of the test-piece. The elongation shall be measured on a standard test-piece having a gauge-length of 8 inches.

For material more than 0.875 inch in thickness the width of the test-piece between the gauge-points shall not exceed 1½ inches; for material less than 0.375 inch thickness the width shall not be more than 2½ inches. In other respects the test-pieces shall conform generally to the standard test-piece *A* (Fig. 111).

Any straightening of test-pieces which may be required shall be done cold.

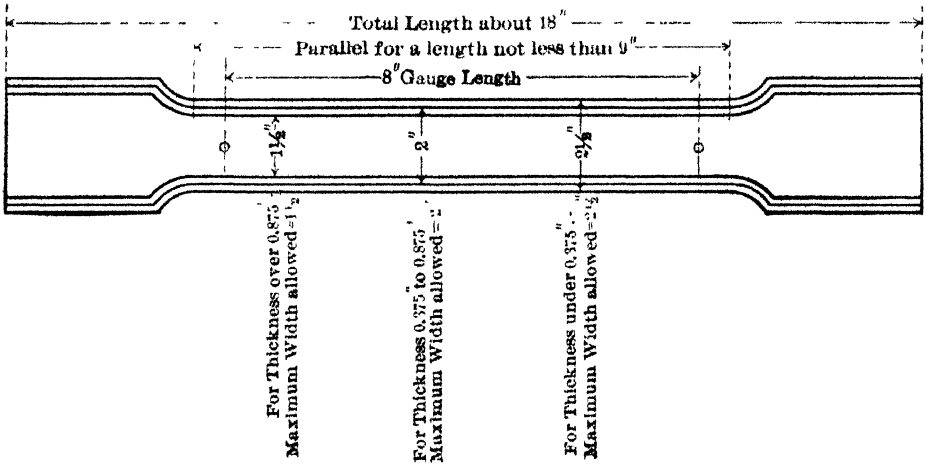
Total Length about 18"
Parallel for a length not less than 9"
8" Gauge Length
1½"
2"
2½"
For Thickness over 0.875"
Maximum Width allowed = 1½"
For Thickness 0.375" to 0.875"
Maximum Width allowed = 2"
For Thickness under 0.375"
Maximum Width allowed = 2½"

FIG. 111.

5. *Mechanical Tests and Selection of Test-pieces.* — Plates and bars for shipbuilding shall comply with the following mechanical tests. All test-pieces shall be selected by the Surveyor and tested in his presence and he shall satisfy himself that the conditions herein described are fulfilled.

6. *Tensile Tests.* — *Plates.* — The tensile breaking strength of steel plates, determined from standard test-pieces, shall be between the limits of 28 and 32 tons (of 2240 pounds each) per square inch. For plates specially intended for cold flanging and marked for identification the lower limit shall be 26 tons per square inch. In the case of material for purposes in which tensile strength is not

important, the tensile test may be dispensed with and the bend-test only be made, if so specified by the builders and approved by the Committee. The elongation, measured on a standard test-piece having a gauge-length of 8 inches, shall not be less than 20 per cent for material of 0.375 inch in thickness and upwards, and not less than 16 per cent for material below 0.375 inch in thickness.

Angles, Bulb Angles, Channels, etc. — The tensile breaking strength of sectional material, such as angles, bulb angles, channels, etc., shall be between the limits of 28 and 33 tons (of 2240 pounds each) per square inch. In the case of material for purposes in which tensile strength is not important, the tensile test may be dispensed with and the bend-test only be made, if so specified by the builders and approved by the Committee. The elongation measured on a standard test-piece having a gauge-length of 8 inches shall not be less than 20 per cent for material of 0.375 inch in thickness and upwards, and not less than 16 per cent for material below 0.375 inch in thickness.

Rivet Bars. — The tensile breaking strength of rivet bars, when required by the Committee to be tested, shall be between the limits of 25 and 30 tons (2240 pounds) per square inch of section, with an elongation of not less than 25 per cent of the gauge-length of eight times the diameter of the test-piece, measured on the standard test-piece *B* (Fig 112). The bars may be tested the full size as rolled.

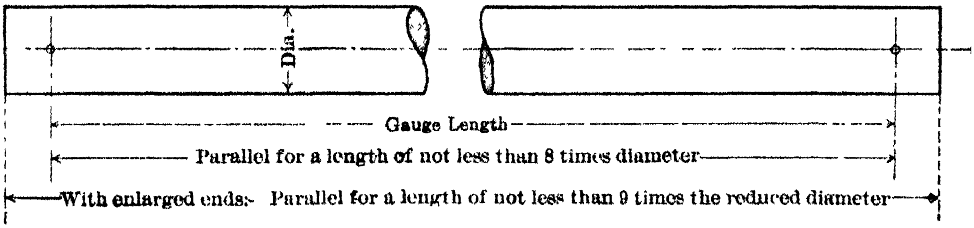
Dia.
Gauge Length
Parallel for a length of not less than 8 times diameter
With enlarged ends:- Parallel for a length of not less than 9 times the reduced diameter

FIG. 112.

When the Surveyor is in constant attendance at the steel works the following requirements are to be complied with:

7. *Number of Tensile Tests.* — *Plates and Sectional Material.* — One tensile test for plates or sectional material shall be taken from the finished material of each charge.

When the quantity of the material from one charge exceeds 25 tons, a second tensile test will be required, also additional tests shall be made for every variation in thickness of 0.15 of an inch in the plates or sectional bars from each charge.

Rivet Bars. — When required by the Committee, one tensile test shall be taken from each charge used for rivet bars; but when the weight of the bars, as rolled, from one charge exceeds 10 tons, an additional tensile test shall be made for each further 10 tons or portion thereof.

Should a tensile-test piece break outside the middle half of its gauge-length, and the elongation be less than that required by the Rules, the test may, at the maker's option, be discarded and another test be made of the same plate or bar.

8. *Bend-tests. — Cold Bends.* — Test-pieces shall be sheared lengthwise or crosswise from plates or bars, and shall not be less than 1½ inches wide, but for small bars the whole section may be used. For rivet bars bend-tests are not required.

Temper Bends. — The test-pieces shall be similar to those used for cold-bend tests. For temper-bend tests the samples shall be heated to a blood-red and quenched in water at a temperature not exceeding 80 degrees Fahr. The color shall be judged indoors in the shade.

In all cold-bend tests, and in temper-bend tests on samples 0.5 inch in thickness and above, the rough edge or arris caused by shearing may be removed by filing or grinding, and samples 1 inch in thickness and above may have the edges machined, but the test-pieces shall receive no other preparation. The test-pieces shall not be annealed unless the material from which they are cut is similarly annealed, in which case the test-pieces shall be similarly and simultaneously treated with the material before testing.

For both cold and temper bends the test-piece shall withstand, without fracture, being doubled over until the internal radius is equal to 1½ times the thickness of the test-piece, and the sides are parallel.

For small sectional material these bend-tests may be made from a flattened bar.

Bend-tests may be made either by pressure or by blows.

9 *Number of Bend-tests.* — A cold or temper-bend test shall be made from each plate or bar as rolled, and these tests shall be in about equal numbers from each charge; but a cold-bend test shall be made from all plates which are specially marked for cold flanging

10. *Tests for Manufactured Rivets.* — Rivets selected by the Surveyor from the bulk shall withstand the following tests:

(*a*) The rivet shanks are to be bent cold, and hammered until the two parts of the shank touch in the manner shown in Fig. 113, without fracture on the outside of the bend

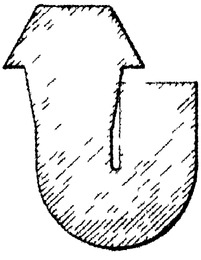

FIG 113

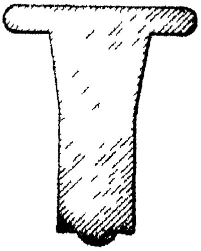

FIG. 114

(*b*) The rivet heads are to be flattened, while hot, in the manner shown in Fig. 114, without cracking at the edges. The heads are to be flattened until their diameter is 2½ times the diameter of the shank.

11. *Additional Tests before Rejection.* — Should any of the test-pieces first selected by the Surveyor not fulfil the test requirements, two further tests may be

made from the same plate or bar, but should either of these fail, the plate or bar from which the test-pieces were cut shall be rejected. In all such cases further tests shall be made before any material from the same charge can be accepted.

12. *Branding.* — Every plate and bar shall be clearly and distinctly marked by the maker in two places with the Society's brand, indicating that the material has complied with the Society's tests.

No plates or bars bearing this brand shall be forwarded from the steel works until the prescribed tests have been made by the Surveyor, and the mill sheets have been signed by him. All plates and bars shall also be legibly stamped in two places with the maker's name or trade-mark and the place where made. They shall also be stamped with numbers or identification marks by which they can be traced to the charge from which the material was made.

13. *Maker's Certificate.* — Before the mill sheets are signed by the Surveyor, the maker shall furnish him with a certificate guaranteeing that the material has been made by the open-hearth process, and that it has been subjected to, and withstood satisfactorily the tests above described in the presence of the Surveyor. The following form of certificate will be accepted if printed on each mill sheet with the name of the firm, and initialed by the test-house manager:

"We hereby certify that the material described below has been made by the open-hearth process, and is that which has been satisfactorily tested in the presence of the Surveyor in accordance with the Rules of Lloyd's Register."

14. *Rejected Material.* — In the event of the material failing in any case to withstand the prescribed tests, the Surveyor shall see that the Society's brand stamped on the plates and bars by the maker has been defaced by punch marks extending beyond the brand in the form of a cross, denoting that the material has been rejected.

AMERICAN FOUNDRYMEN'S ASSOCIATION STANDARD SPECIFICATIONS FOR TESTING GRAY CAST IRON.

(June 4, 1901.)

1. Unless furnace iron, dry sand or loam moulding, or subsequent annealing is specified, all gray-iron castings are understood to be of cupola metal; mixtures, moulds and methods of preparation to be fixed by the founder to secure the results desired by purchaser.

2. All castings shall be clean, free from flaws, cracks, and excessive shrinkage. They shall conform in other respects to whatever points may be specially agreed upon.

3. When the castings themselves are to be tested to destruction, the number selected from a given lot and the tests they shall be subjected to are made a matter of special agreement between founder and purchaser.

4. Castings made under these specifications, the iron in which is to be tested for its quality, shall be represented by at least three test-bars from the same heat.

5. These test-bars shall be subjected to a transverse-breaking test, the load applied at the middle with supports 12 inches apart. The breaking-load and deflection shall be agreed upon specially on placing the contract, and two of these bars shall meet the requirements.

6. A tensile-strength test may be added, in which case at least three bars for this purpose shall be cast with the others in the same moulds respectively. The ultimate strength shall also be agreed upon specially before placing the contract, and two of the bars shall meet the requirements.* (See Fig. 115 for shape of tension-test piece.)

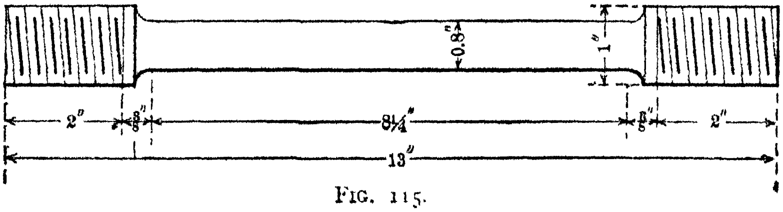

FIG. 115.

7. The dimensions of the test-bars shall be as given herewith. There is only one size for the tensile bar and three for the transverse. For the light and medium weight castings the $1\frac{1}{4}$-inch round bar is to be used, heavy castings the 2-inch, and chilling irons the $2\frac{1}{2}$-inch test-bar.

8. Where the chemical composition of the castings is a matter of specification in addition to the physical tests, borings shall be taken from all the test-bars made, well mixed, and any required determination, combined carbon and graphite alone excepted, made therefrom.†

9. Reasonable facilities shall be given the inspectors to satisfy themselves that castings are being made in accordance with specifications, and, if possible, tests shall be made at the place of production prior to shipments.

10. The following are the sizes of bars selected for tests as a result of our investigations:

For all tension tests a bar turned to 0.8 inch in diameter, corresponding to a cross-section of $\frac{1}{2}$ square inch. (Fig. 115.) Results, therefore, multiplied by two give the tensile strength per square inch.

For transverse test of all classes of iron for general comparison, a bar $1\frac{1}{4}$ inches diameter on supports 12 inches apart, pressure applied in middle and deflection noted. Similarly, for ingot-mould, light machinery, stove-plate and novelty iron a 1½-inch diameter bar; that is to say, for irons running from two per cent in silicon upward, or from 1.75 per cent silicon upward where but little scrap is in the mixture.

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* The remarkably wide range of values for the ultimate strength and modulus of rupture which are really good for the various classes of iron precludes the giving of definite upper limits in the specifications. It will therefore remain a matter of mutual agreement in each case, the requirements of service and price per pound paid regulating the mixtures which can be used.

† There should really be no necessity for this test, for the requirements of the physical tests presuppose a given chemical composition. It may, however, sometimes be expedient to know the total carbon, silicon, sulphur, manganese, and phosphorus of a casting to insure good service conditions.

For dynamo frame, sash weight, cylinder, heavy machinery, and gun-metal irons, similarly a 2-inch diameter bar is recommended; that is, for irons running from 1.50 to 2 per cent in silicon, or where the silicon is lower and the proportion of scrap is rather large

For roll irons, whether chilled or sand, and car-wheel metals, a 2½-inch diameter bar is recommended; that is, for all irons below 1 per cent silicon, and which may therefore be classed as the chilling irons.

AMERICAN WATER-WORKS ASSOCIATION STANDARD SPECIFICATIONS FOR CAST-IRON WATER-PIPE. (Philadelphia, 1891.)

1. *Length* — Each pipe shall be of the kind known as "socket and spigot,' and shall be 12 feet long from bottom of the socket to the end of the pipe.

2. *Metal and Treatment.* — The metal shall be best quality neutral pig-iron, with no admixture of cinder, cast in dry-sand moulds, placed vertically, numbered and marked with name of maker and date of making. The shell to be smooth and round, without imperfections, and of uniform thickness.

All pipes to be thoroughly cooled when taken from the pit, afterward thoroughly cleaned without the use of acid, then heated to 300 degrees F., and plunged into coal-pitch varnish. When removed, the coating to fume freely and set hard within an hour.

Templates to be furnished by the maker; the weight of pipe to vary not over 3 per cent from the standard; all tests to be made at expense of maker.

3. *Testing.* — The pipes to be tested after the varnish hardens with hydrostatic pressure of 300 pounds per square inch for all sizes below 12 inches diameter, and 250 pounds for all above that diameter, and simultaneously to be struck with a 3-pound hammer.

4. *Test-bars.* — Test-bars to be 26 inches long, 2 inches wide, and 1 inch thick, and to be tested for transverse strength. These bars shall stand, when carried flatwise on supports 24 inches apart, a center load of 1900 pounds, and show a deflection of not less than 0.25 inch before breaking. Test-bars are to be cast when required by the inspector, and to be as nearly as possible the specified dimensions.

PENNSYLVANIA RAILROAD COMPANY'S TEST FOR CAST-IRON CAR-WHEELS.

Car-wheels are usually subjected to the drop-test. The following method is employed by the Pennsylvania Railroad Company for testing cast-iron wheels:

For each fifty wheels which have been shipped, or are ready to ship, one wheel is taken at random by the railroad company's inspector, either at the railroad company's shops or at the wheel-manufacturer's, as the case may be, and subjected to the following test: The wheel is placed flange downward on an anvil-block weighing 1700 pounds, set on rubble masonry two feet down, and having three supports not more than five inches wide for the wheel to rest on. This arrangement being effected, the wheel is struck centrally on the hub by a weight of 140 pounds, falling from a height of 12 feet. Should the wheel break in two or more pieces before nine blows or less, the fifty wheels represented by it are rejected. If the wheel stands eight blows without breaking in two or more pieces, the fifty wheels are accepted.

82. Tests of Building Stone and Brick. — These materials are principally used in walls of buildings and for foundations. For this use they are subjected principally to compression or crushing stresses. The important properties are strength and durability. Stone is usually tested for compressive and transverse strength, brick for compressive strength.

1. *Testing Stones.* —

The specimens for compressive strength are cubes of various sizes, depending principally on the capacity of the testing-machine. These cubes are to be nicely made with the opposite sides perfectly parallel to provide a uniform bearing-surface. It is found that the larger the blocks the greater the strength per unit of area.*

To test Stone for Compressive Strength. — Have the specimen dry and dressed, and ground to a cube — inches on each edge, and with the opposite faces parallel planes. This is important, as imperfect or wedge-shaped faces concentrate the stress on a small area. In testing, use a layer of wet plaster of Paris between the specimen and the faces of the machine, to distribute the stress.

To test Stone for Transverse Strength. — In this case the specimen is dressed into the form of a prism 8 inches long and 2 by 2 inches in section. It is supported on bearings 6 inches apart, and a center load applied. The strength is computed as explained under head of Transverse Testing, page 60.

Durability of stone is tested accurately only by actual trial. Some idea can be formed by noticing the effect of the weather on the exposed rocks in the quarry from which the specimen came.

In the method of standard tests adopted in Munich, in 1887, the following additional tests are recommended:

Trial method with (*a*) a jumper or drill, (*b*) by rotary boring. The amount of work done by the drill to be determined by the momentum of drop, its velocity of rotation, and the shape or cutting angle of the drill or cutting tool. These qualities are to be determined by comparison with a standard drill working under definite conditions. Examine the stone for resistance to shearing as well as to boring.

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* See Unwin, "Testing of Materials."

Find when possible the position in the quarry originally occupied by the specimen tested.

Find out the intended use of the stone, and determine the character of tests largely from that. Dry the stone until no further loss of weight occurs at a temperature of 30° C. (86° F.), and test in a dry condition.

Make the tests for strength as described, using as large specimens as possible. Also, test by compression rectangular blocks. Test also for tension and bending.

Obtain the specific gravity, after drying at a temperature of 86° F.

Examine the specimen for *resistance to frost* by using samples of uniform size, 7 cm. (2.76 inches) on each edge.

The *frost-test* consists of:

a. The determination of the *compressive strength* of *saturated* stones, and its comparison with that of dried pieces.

b. The determination of *compressive strength* of the dried stone after having been frozen and thawed out twenty-five times, and its comparison with that of dried pieces not so treated.

c. The determination of the loss of weight of the stone after the twenty-fifth frost and thaw. Special attention must be had to the loss of those particles which are detached by the *mechanical* action, and also those lost by solution in a definite quantity of water.

d. The examination of the frozen stone by use of a magnifying-glass, to determine particularly whether fissures or scaling occurred.

For the frost-test are to be used:

Six pieces for compression-tests in dry condition, three normal and three parallel to the bed of the stone, provided these tests have not already been made, in which case it is permissible, on account of the law of proportions, to use cubical test-blocks larger than 7 cm. (2.76 inches).

Six test-pieces in saturated condition — not frozen, however; three tested normal to and three parallel to bed.

Six test-pieces for tests when frozen, three of which are to be tested normal to and three parallel to bed of stone.

When making the freezing-test the following details are to be observed:

a. During the absorption of water the cubes are at first to be immersed but 2 cm. (0.77 inch) deep, and are to be lowered little by little until finally submerged.

b. For immersion, distilled water is to be used at a temperature of from 15° C. (59° F.) to 20° C. (68° F.).

c. The saturated blocks are to be subjected to temperatures of from − 10° to − 15° C. (14° to 5° F.). This can be done in a vessel surrounded with melting ice and salt.

d. The blocks are to be subjected to the influence of such cold for four hours, and they are to be thus treated when completely saturated.

e. The blocks are to be thawed out in a *given quantity* of distilled water at from 59° F. to 64° F.

An investigation of *weathering* qualities — stability under influences of atmospheric changes — can be neglected when the frost-test has been made. However, the effects in this respect, in *nature*, are to be carefully observed and compared with previous experience in the use of similar material. Observe —

a. The effect of the sun in producing cracks and ruptures in stones.
b. The effect of the air, and whether carbonic-acid gas is given off.
c. The effect of rain and moisture.
d. The effect of temperature

2. *Testing Bricks or Artificial Building stone.* — Bricks are tested for strength, principally by compression.

They should be ground to a form with opposite parallel faces, and are tested between layers of thin paper, or, without grinding, between thin layers of plaster of Paris, as explained for stone. The variation in size of specimen, and whether the brick is tested on end, side-ways, or flat-ways, will make a great difference in the results. The test, to be of any value, must state the method of testing. Whole bricks are stronger per unit of area than portions of bricks, and should be used when practicable.

It is also recommended that brick be tested for compression in the shape of two half-bricks superimposed, united by a thin layer of Portland cement, and covered on top and bottom with a thin layer of such paste to secure even bearing-surfaces.*

The transverse test for brick is believed to be a valuable index to its building properties. Support the brick on knife-edges 6 inches apart, and apply the load at the center. Compute the modulus of rupture:

$$R = \frac{3}{2}\frac{Wl}{bd^2},$$

in which W equals the center-load, l the length, b the breadth, d the depth, all in inches.

Dry as for stone, and determine the *specific gravity*.

Test hard-burned and soft-burned from the same kiln

Determine the *porosity* of the brick as follows·

Thoroughly dry ten pieces on an iron plate; weigh these pieces; then submerge in water to one-half the depth for twenty-four hours; then completely submerge for twenty-four hours, dry superficially, and weigh. Determine porosity from the weight of water absorbed, which should be expressed as per cent of volume. Express *absorption* as per cent of weight.

Determine resistance against frost, as previously explained for stones, using five specimens, and repeating the operation of freezing and thawing twenty-five times for each specimen. Observe the effect with a magnifying-glass. After freezing, test for compression, and compare the results with those obtained with a dry brick.

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* See Vol. XI (Standard Method of Testing), Transactions of American Society Mechanical Engineers.

To test brick for *soluble salts*, obtain samples from an underburned brick and grind these to dust. Sift through a sieve 4900 meshes per square cm. (31,360 per square inch). The dust sifted out is lixiviated in 250 c.c. of distilled water, boiled for about one hour, filtered, and washed. The amount of soluble salts is then determined by boiling down the solution and bringing the residue to a red heat for a short time. The amount is determined by weight and expressed in percentage; its composition is determined by a chemical analysis.

Determinations of the presence of carbonate of lime, mica, or pyrites are to be made by chemical analysis.

83. Tests of Paving Material, Stones, and Ballast, Natural and Artificial.

— In this case the following observations and tests should be made:

Information in regard to petrographic and geologic *classification*, the *origin* of the samples, etc., etc.; also:

Statement in regard to *utilization* of same.

Specific gravity of the samples is to be determined.

All materials used in the construction of roads, provided they are not to be used under cover or in localities without frost, are to be tested for their *frost-resisting qualities* by tests similar to those prescribed for natural stone.

Stones or brick used for paving are tested most satisfactorily in a manner representing their mode of utilization by determining the *wearing qualities* by an abrasion-test standardized by the National Brick Manufacturers' Association (1900) as follows:

The Rattler Test. — The standard rattler shall be 28 inches in diameter and 20 inches in length, inside measurements. Other dimensions may be employed between 26 and 30 inches diameter and 18 to 24 inches length, in which case the dimensions should be stated in reporting the test Longer rattlers may be employed by the insertion of a diaphragm

The barrel shall be supported on trunnions at the ends with no shaft running through the rattling chamber. The cross-section shall be a regular polygon of 14 sides. The heads shall be of gray cast iron, not chilled or case hardened. The staves shall preferably be composed of steel plates, as cast-iron peens and ultimately breaks from the wearing action on the inner side There shall be a space of one-fourth of an inch between the staves for the escape of dust and small pieces. Machines having from 12 to 16 staves may be employed, with openings from ⅛ to ⅜ inch, but these variations from the standard should be mentioned in an official report.

The *Charge* shall consist of but one kind of brick at a time, nine paving blocks or twelve bricks* being inserted, together with 300 pounds of cast-iron blocks.

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* The number of bricks should be that number which most nearly gives the total cubic contents of the brick charge equal to 8 per cent of the total cubic contents of the rattler.

These shall be of two sizes, 75 pounds being of the larger and 225 pounds of the smaller size. The larger size shall be about $2\frac{1}{2}$ inches square and $4\frac{1}{2}$ inches long, with slightly rounded edges, and shall weigh at first $7\frac{1}{2}$ pounds. The smaller size shall be $1\frac{1}{2}$-inch cubes, with rounded edges. All blocks shall be replaced by new ones when they have lost 10 per cent of their normal weight.

The *Number of Revolutions* shall be 1800 for a standard test, at a speed between 28 and 30 per minute.

The Bricks shall be thoroughly dried before testing.

The Loss shall be calculated as a per cent of the weight of the dry bricks composing the charge, and *no result shall be considered as official unless it is the average of two distinct and complete tests on separate charges of bricks.*

The uniformity of wearing qualities of brick for parts more or less distant from the exterior surface is determined by repeating the trial on the same piece, and not merely testing *one*, but a *greater number* of pieces. It is, moreover, necessary to test samples of the best, the poorest, and the medium qualities of bricks in any one kiln.

Obtain the *transverse strength* as explained.

Obtain the *per cent of water absorbed* after the bricks have been thoroughly dried at 30° C. (86° F.), as explained in Art. 2, p. 154.

Test materials for ballast in a similar manner.

In some cases it may be desirable to test stones as to the capacity for receiving a polish.

Examinations of *asphalts* can only be made in an exhaustive manner by the construction of trial roads. An opinion coinciding with the results of such trial may be formed by —

(*a*) Determination of the quantity and quality of the bitumen contained therein (whether the bitumen be artificial or natural).

(*b*) By physical and chemical determination of the residue.

(*c*) By determination of the specific density of test-pieces of the material used by means of a needle of a circular sectional area of 1 sq. mm., carrying a weight of 300 grams. (See Cement Testing, p. 160.)

(*d*) By the determination of the wear of such test-pieces by abrasion or grinding trials.

(*e*) By the determination of the resistance to frost of these test-pieces. (See Building Stone, Frost-test.)

84. Testing Cements and Mortars. — The following descriptions will serve to distinguish the different classes of bonding materials:

1. *Common limes* are produced by roasting at bright red heat limestones containing little clay or silicic acid. When moistened with water limes become wholly or partly pulverized and slaked. They are sold, when unslaked, as lumps of "quick-lime"; when slaked, in the form of fine flour. After mixing with water (and sand) to

make mortar, the setting or hardening of the lime cement takes place by chemical combination with CO_2 from the atmosphere, and simultaneous rejection and evaporation of the water used in mixing.

2. *Water limes* and *Roman cements* are products obtained by burning clayey lime marls or limestones below the melting temperature. They do not disintegrate on being moistened, but must be powdered by mechanical means. Their setting or hardening is similar to that of the hydraulic cements rather than to that of common lime.

3. *Hydraulic cements* are products obtained by burning at a temperature of incipient fusion clayey marls or limestones or artificial mixtures of materials containing clay and lime (or lime, silica, and alumina), the clinker so made being crushed and ground to the fineness of flour. The hydraulic parts of a hydraulic cement are silicates and aluminates (or ferrates) of lime. In setting or hardening of the cement the hydraulic parts combine chemically with part or all of the water added in mixing; the water disappears as water, and becomes part of the cement. Hence the name "hydraulic cement."

4. *Hydraulic fluxes* are natural or artificial materials which in general do not harden of themselves, but do so in presence of caustic lime, and then in the same way as a hydraulic material; i.e., puzzuolana, santorine earth, trass produced from a proper kind of volcanic tufa, blast-furnace slag, burnt clay.

5. *Puzzuolana cements* are products obtained by most carefully mixing hydrates of lime, pulverized, with hydraulic fluxes in the condition of dust.

6. *Mixed cements* are products obtained by most carefully mixing existing cements with proper fluxes. Such bond materials should be particularly marked "Mixed Cements," at the same time naming the base and the flux used.

7. *Mortar* is made by mixing three or four parts of sharp sand with one part of quick-lime or cement, and adding water until of the proper consistency. Mortar made from *quick-lime* will neither set nor stay hard under water; that made from *hydraulic-* or *water-lime*, if allowed to set in the air, will not be softened by water; while that made from hydraulic cements will harden under water.

8. *Method of Testing Cements.* — The principal properties which it is necessary to know are: (1) fineness; (2) time of setting; (3) tensile strength; (4) soundness or freedom from cracks after setting; (5) heaviness or specific gravity; (6) crushing strength; (7) toughness or power to resist definite blows.

The following standard method of testing cements was adopted by a committee of the American Society of Civil Engineers and of the American Society for Testing Materials in 1903 and 1904.

Selection of Sample. — The sample shall be a fair average of the contents of the package; it shall be passed through a sieve having 20 meshes per lineal inch before testing to remove lumps. In obtaining a sample from barrels or bags, an auger or sampling-iron reaching to the center should be used.

A chemical analysis, if required, should be made in accordance with the directions in the Journal of the Society of Chemical Industry, published Jan. 15, 1902.

Specific Gravity. — This is most conveniently made with le Chatelier's apparatus, which consists of a flask (*D*), Fig. 116, of 120 cu. cm. (7.32 cubic inches) capacity, the neck of which is about 20 cm. (7.87 inches) long; in the middle of this neck is a bulb (*C*), above and below which are two marks (*F* and *E*); the volume between these marks is 20 cu. cm. (1.22 cubic inches). The neck has a diameter of about 9 mm. (0.35 in.), and is graduated into tenths of cubic centimeters above the mark *F*. Benzine (62° Baumé naphtha), or kerosene free from water, should be used in making the determination.

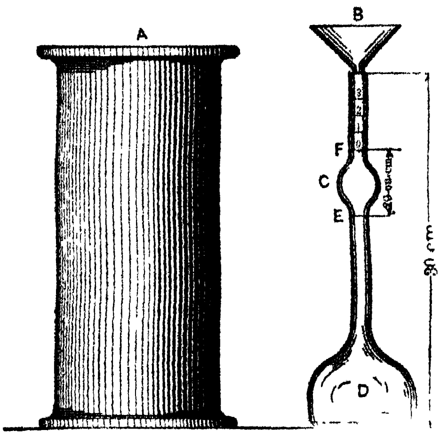


FIG. 116. — LE CHATELIER'S SPECIFIC-GRAVITY APPARATUS.

The specific gravity can be determined in two ways: (1) The flask is filled with either of these liquids to the lower mark (*E*), and 64 gr. (2.25 ounces) of powder, previously dried at 100° C. (212° F.) and cooled to the temperature of the liquid, is gradually introduced through the funnel (*B*) [the stem of which extends into the flask to the top of the bulb (*C*)], until the upper mark (*F*) is reached. The difference in weight between the cement remaining and the original quantity (64 gr.) is the weight which has displaced 20 cu. cm.

(2) The whole quantity of the powder is introduced, and the level of the liquid rises to some division of the graduated neck. This reading plus 20 cu. cm. is the

volume displaced by 64 gr. of the powder. The specific gravity is then obtained from the formula:

$$\text{Specific gravity} = \frac{\text{Weight of cement}}{\text{Weight of volume of liquid displaced}}.$$

The flask during the operation is kept immersed in water in a jar, *A*, in order to avoid variations in the temperature of the liquid. Different trials should agree within 1 per cent.

The apparatus is conveniently cleaned by inverting the flask over a glass jar, then shaking it vertically until the liquid starts to flow freely. Repeat this operation several times.

Fineness. — The fineness is determined by the use of circular sieves, about 20 cm. (7.87 inches) in diameter, 6 cm. (2.36 inches) high, and provided with a pan 5 cm. (1.97 inches deep) and a cover.

The wire cloth should be woven (not twilled) from brass wire having the following diameters:

No. 100, 0.0045 inch; No. 200, 0.0024 inch.

This cloth should be mounted on the frames without distortion; the mesh should be regular in spacing and be within the following limits:

No. 100, 96 to 100 meshes to the linear inch;
No. 200, 188 to 200 " " " " "

50 to 100 gr. dried at a temperature of 212° F. prior to sieving should be used for the test, the sieves having previously been dried.

The coarsely screened sample is weighed and placed on the No. 200 sieve, which is moved forward and backward, at the same time striking the side gently with the palm of the other hand, at the rate of about 200 strokes per minute. The operation is continued until not more than one-tenth of one per cent passes through per minute. The work is expedited by placing in the sieve a small quantity of large shot, or, better, some flat pieces of brass or copper about the size of a cent. The residue is weighed, then placed on a No. 100 sieve and the operation repeated. The results should be reported to the nearest tenth of one per cent.

Normal Consistency. — The use of a proper percentage of water in mixing the cement or mortar is exceedingly important. No method is entirely satisfactory, but the following, which consists in the determination of the depth of penetration of a wire of a known diameter carrying a specified weight, is recommended. The apparatus recommended is the *Vicat needle*, shown in Fig. 117, which is also used for determining the time of setting. This consists of a frame, *K*, bearing a movable rod, *L*, with a cap, *D*, at one end, and at the other the cylinder, *G*, 1 cm. (0.39 inch) in diameter, the cap, rod and cylinder weighing 300 gr. (10.58 oz.) The rod, which can be held in any desired position by a screw, *F*, carries an indicator, which moves over a graduated scale attached to the frame, *K*. The paste is held by a conical hard-rubber ring, *I*, 7 cm. (2.76 inches) in diameter at the base, 4 cm. (1.57 inches) high, resting on a glass plate, *J*, about 10 cm. (3 94 inches) square.

In making the determination, the same quantity of cement as will be subsequently used for each batch in making the briquettes (but not less than 500 grams) is kneaded into a paste and quickly formed into a ball with the hands, completing the operation by tossing it six times from one hand to the other, maintained 6 inches apart; the ball is then pressed into the rubber ring, through the larger opening, smoothed off, and placed (on its large end) on a glass plate and the smaller end smoothed off with a trowel; the paste, confined in the ring, resting on the plate, is placed under the rod bearing the cylinder, which is brought in contact with the surface and quickly released.

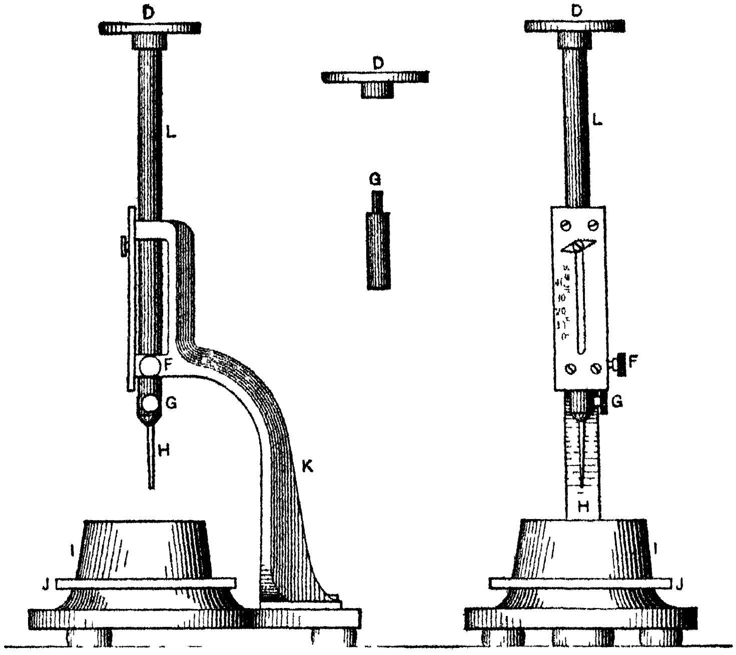


FIG. 117. — VICAT NEEDLE.

The paste is of normal consistency when the cylinder, released from contact with the surface, penetrates to a point in the mass 10 mm. (0.39 inch) below the top of the ring and there stops. Great care must be taken to fill the ring exactly to the top.

The trial pastes are made with varying percentages of water until the correct consistency is obtained.

The committee has recommended, as normal, a paste the consistency of which is rather wet, because it believes that variations in the amount of compression to which the briquette is subjected in moulding are likely to be less with such a paste.

Time of Setting. — The object of this test is to determine the time which elapses until the paste ceases to be fluid and plastic, called the initial set, and also the time required for it to acquire a certain degree of hardness, called the final set.

For this purpose the Vicat needle, which has already been described, should be used. In making the test, a paste of normal consistency is moulded and placed under the rod (*L*), Fig. 117; this rod when bearing the cap (*D*) weighs 300 gr. (10.58 oz.). The needle (*H*), at the lower end, is 1 mm. (0.039 inch) in diameter. Then the needle is carefully brought in contact with the surface of the paste and quickly released.

The setting is said to have commenced when the needle ceases to pass a point 5 mm. (0.20 inch) above the upper surface of the glass plate, and is said to have terminated the moment the needle does not sink visibly into the mass.

The test-pieces should be stored in moist air during the test. This is accomplished by placing them in a rack over water contained in a pan and covered with a damp cloth, the cloth to be kept away from them by means of a wire screen, or preferably they may be stored in a moist box or closet.

The determination of the time of setting is only approximate, since it is materially affected by the temperature of the mixing water, the percentage of the water used, and the amount of moulding the paste receives.

Standard Sand. — The committee recommend at present the use of a natural sand from Ottawa, Ill., screened to pass a sieve having 20 meshes per lineal inch and retained on a sieve having 30 meshes per lineal inch; the wires to have diameters of 0.0165 and 0.0112 inch respectively. This sand will be furnished by the Sandusky Portland Cement Co., Sandusky, Ohio, at a moderate price. This sand gives in testing considerably more strength than the crushed quartz of the same size formerly employed for this purpose.

Form of Briquette. — The form of briquette recommended is shown in Fig. 120. It is substantially like that formerly used except that the corners are rounded.

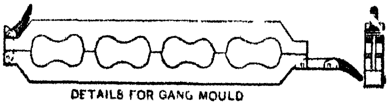
Moulds. — The moulds should be made of brass, bronze, or some equally non-corrodible material, and gang moulds of the form shown in Fig. 118 are recommended. They should be wiped with an oily cloth before using.

Mixing. — All proportions should be stated by weight; the quantity of water to be used should be stated *as a percentage of the dry material.* The metric system is recommended because of the convenient relation of the gram and the cubic centimeter. The temperature of the room and the mixing water should be as near 21° C. (70° F.) as it is practicable to maintain it.

The sand and cement should be thoroughly mixed dry. The mixing should be done on some non-absorbing surface, preferably plate glass. If the mixing must be done on an absorbing surface, it should be thoroughly dampened prior to use. The quantity of material to be mixed at one time depends on the number of test-pieces to be made; about 1000 gr. (35.28 oz.) makes a convenient quantity to mix, especially by hand methods.

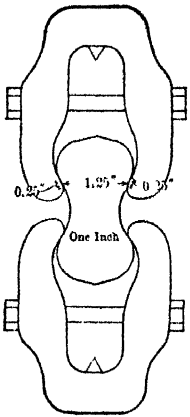
The material is weighed, dampened, and roughly mixed with a trowel, after which the operation is completed by vigorously kneading with the hand for 1½ minutes.

Moulding. — Having worked the mortar to the proper consistency it is at once placed in the mould by hand, being pressed in firmly with the fingers and smoothed off with a trowel without ramming, but in such a manner as to exert a moderate pressure. The mould should be turned over and the operation repeated. The briquettes should be weighed prior to immersion, and those which vary in weight more than 3 per cent from the average should be rejected.



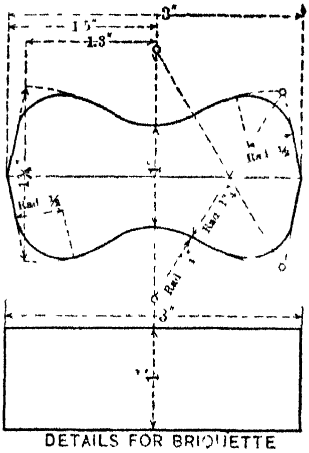
DETAILS FOR GANG MOULD

FIG. 118.



FORM OF CLIP

FIG. 119.



DETAILS FOR BRIQUETTE

FIG. 120.

STANDARD CLIP AND BRIQUETTE ADOPTED BY THE AMERICAN SOCIETY FOR TESTING MATERIALS, 1904.

Storage of the Test-pieces.—During the first twenty-four hours after moulding, the test-pieces should be kept in moist air to prevent them from drying out. A moist closet or chamber is so easily devised that the use of the damp cloth should be abandoned if possible. Covering the test-pieces with a damp cloth is objectionable, as commonly used, because the cloth may dry out unequally, and, in consequence, the test-pieces are not all maintained under the same condition. Where a moist closet is not available, a cloth may be used and kept uniformly wet by immersing the ends in water. It should be kept from direct contact with the test-pieces by means of a wire screen or some similar arrangement.

A moist closet consists of a soapstone or slate box, or a metal-lined wooden box —the metal lining being covered with felt and this felt kept wet. The bottom of the box is so constructed as to hold water, and the sides are provided with cleats for holding glass shelves on which to place the briquettes. Care should be taken to keep the air in the closet uniformly moist.

After twenty-four hours in moist air the test-pieces for longer periods of time should be immersed in water maintained as near 21° C. (70° F.) as practicable; they may be stored in tanks or pans, which should be of non-corrodible material.

Tensile Strength. — The tests may be made on any standard machine. A solid metal clip, as shown in Fig. 119, is recommended. This clip is to be used without cushioning at the points of contact with the test-specimen. The bearing at each point of contact should be ¼ inch wide, and the distance between the center of contact on the same clip should be 1¼ inches.

Test-pieces should be broken as soon as they are removed from the water, the load being applied uniformly at the rate of about 600 pounds per minute. The average tests of the briquettes of each sample should be taken as the strength, excluding any results which are manifestly faulty.

Constancy of Volume. — The object is to develop those qualities which tend to destroy the strength and durability of a cement. As it is highly essential to determine such qualities at once, tests of this character are for the most part made in a very short time, and are known, therefore, as accelerated tests. Failure is revealed by cracking, checking, swelling, or disintegration, or all of these phenomena. A cement which remains perfectly sound is said to be of constant volume.

Tests for constancy of volume are divided into two classes: (1) normal tests, or those made in either air or water maintained at about 21° C. (70° F.), and (2) accelerated tests, or those made in air, steam, or water at a temperature of 45° C. (115° F.) and upward. The test-pieces should be allowed to remain twenty-four hours in moist air before immersion in water or steam, or preservation in air.

For these tests, pats, about 7½ cm. (2.95 inches) in diameter, 1¼ cm. (0.49 inch) thick at the center, and tapering to a thin edge, should be made, upon a clean glass plate [about 10 cm (3.94 inches) square], from cement paste of normal consistency.

Normal Test. — A pat is immersed in water maintained as near 21° C. (70° F.) as possible for 28 days, and observed at intervals. A similar pat is maintained in air at ordinary temperature and observed at intervals.

Accelerated Test — A pat is exposed in any convenient way in an atmosphere of steam, above boiling water, in a loosely closed vessel for three hours.

To pass these tests satisfactorily, the pats should remain firm and hard, and show no signs of cracking, distortion, or disintegration. Should the pat leave the plate, distortion may be detected best with a straight-edge applied to the surface which was in contact with the plate. In the present state of our knowledge it cannot be said that cement should necessarily be condemned simply for failure to pass the accelerated tests, nor can it be considered entirely satisfactory if it has passed these tests.

AMERICAN SOCIETY FOR TESTING MATERIALS' SPECIFICATIONS FOR CEMENT (Nov. 14, 1904).

1. **General Conditions** — (*a*) All cement shall be inspected.

(*b*) Cement may be inspected either at the place of manufacture or on the work.

(*c*) In order to allow ample time for inspecting and testing, the cement should be stored in a suitable weather-tight building having the floor properly blocked or raised from the ground.

(*d*) The cement shall be stored in such a manner as to permit easy access for proper inspection and identification of each shipment.

(*e*) Every facility shall be provided by the contractor and a period of at least twelve days allowed for the inspection and necessary tests.

(*f*) Cement shall be delivered in suitable packages with the brand and name of manufacturer plainly marked thereon.

(*g*) A bag of cement shall contain 94 pounds of cement net. Each barrel of Portland cement shall contain 4 bags, and each barrel of natural cement shall contain 3 bags of the above net weight.

(*h*) Cement failing to meet the seven-day requirements may be held awaiting the results of the twenty-eight-day tests before rejection.

(*i*) All tests shall be made in accordance with the methods proposed by the Committee on Uniform Tests of Cement of the American Society of Civil Engineers, presented to the Society January 21, 1903, and amended January 20, 1904, with all subsequent amendments thereto.

(*j*) The acceptance or rejection shall be based on the following requirements:

2. Natural Cement. — *Definition.* — This term shall be applied to the finely pulverized product resulting from the calcination of an argillaceous limestone at a temperature only sufficient to drive off the carbonic acid gas.

(*a*) *Specific Gravity.* — The specific gravity of the cement thoroughly dried at 100° C. shall be not less than 2.8.

(*b*) *Fineness.* — It shall leave by weight a residue of not more than 10 per cent on the No. 100 sieve, and 30 per cent on the No. 200.

(*c*) *Time of Setting.* — It shall develop initial set in not less than ten minutes, and hard set in not less than thirty minutes nor more than three hours.

(*d*) *Tensile Strength.* — The minimum requirements for tensile strength for briquettes one inch square in cross-section shall be within the following limits, and shall show no retrogression in strength within the periods specified:*

Age.	Strength.
NEAT CEMENT.	
24 hours in moist air	50–100 lbs.
7 days (1 day in moist air, 6 days in water)	100–200 "
28 days (1 day in moist air, 27 days in water)	200–300 "
ONE PART CEMENT, THREE PARTS STANDARD SAND.	
7 days (1 day in moist air, 6 days in water)	25–75 "
28 days (1 day in moist air, 27 days in water)	75–150 "

(*e*) *Constancy of Volume.* — Pats of neat cement about three inches in diameter, one-half inch thick at center, tapering to a thin edge, shall be kept in moist air for a period of twenty-four hours.

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* For example, the minimum requirement for the twenty-four-hour neat-cement test should be some specified value within the limits of 50 and 100 pounds, and so on for each period stated.

1. A pat is then kept in air at normal temperature.

2. Another is kept in water maintained as near 70° F. as practicable.

These pats are observed at intervals for at least 28 days, and, to satisfactorily pass the tests, should remain firm and hard and show no signs of distortion, checking, cracking or disintegrating.

3. Portland Cement. — *Definition.* — This term is applied to the finely pulverized product resulting from the calcination to incipient fusion of an intimate mixture of properly proportioned argillaceous and calcareous materials, and to which no addition greater than 3 per cent has been made subsequent to calcination.

(*a*) *Specific Gravity.* — The specific gravity of the cement, thoroughly dried at 100° C., shall be not less than 3.10.

(*b*) *Fineness.* — It shall leave by weight a residue of not more than 8 per cent on the No. 100 sieve, and not more than 25 per cent on the No. 200.

(*c*) *Time of Setting.* — It shall develop initial set in not less than thirty minutes, but must develop hard set in not less than one hour nor more than 10 hours.

(*d*) *Tensile Strength.* — The minimum requirements for tensile strength for briquettes one inch square in section shall be within the following limits, and shall show no retrogression in strength within the periods specified:*

Age.	Strength.
NEAT CEMENT.	
24 hours in moist air	150–200 lbs.
7 days (1 day in moist air, 6 days in water)	450–550 "
28 days (1 day in moist air, 27 days in water)	550–650 "
ONE PART CEMENT, THREE PARTS SAND.	
7 days (1 day in moist air, 6 days in water)	150–200 "
28 days (1 day in moist air, 27 days in water)	200–300 "

(*e*) *Constancy of Volume.* — Pats of neat cement about three inches in diameter, one-half inch thick at the center, and tapering to a thin edge, shall be kept in moist air for a period of twenty-four hours.

1. A pat is then kept in air at normal temperature and observed at intervals for at least 28 days.

2. Another pat is kept in water maintained as near 70° F. as practicable, and observed at intervals for at least 28 days.

3. A third pat is exposed in any convenient way in an atmosphere of steam, above boiling water, in a loosely closed vessel for five hours.

These pats, to satisfactorily pass the requirements, shall remain firm and hard and show no signs of distortion, checking, cracking, or disintegrating.

(*f*) *Sulphuric Acid and Magnesia.* — The cement shall not contain more than 1.75 per cent of anhydrous sulphuric acid (SO_3), nor more than 4 per cent of magnesia (MgO).

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* For example, the minimum requirement for the twenty-four-hour neat-cement test should be some specified value within the limits of 150 and 200 pounds, and so on for each period stated.

Report Blank. — The following is a blank for reporting the results of cement testing:

DEPARTMENT OF EXPERIMENTAL ENGINEERING, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Cement Test.

Test of Sample of Cement representing .... barrels.
Received from .... Tested by ....
Sampled by ....
Marked ....
Date ....

Laboratory number.	Specific gravity.	Per cent wt. of water / wt. of dry material used in		Temperature at time of mixing of		Fineness, per cent, passing sieve of nominal mesh			Setting time, from adding of water in mixing, to finding of		Tensile strength at ages in days of						Soundness test.		Remarks.
											Neat cement.				1.3 Mortar.				
		Normal consistency paste.	1.3 Sand briquettes.	Air.	Water.	50	100	200	Initial set.	Final set.	1	2	7	28	7	28	Hours steamed.	Condition found.	

CHAPTER VI.

MEASUREMENT OF PRESSURE.

85. **Pressure, Definitions and Units.**—The term *pressure*, as employed in engineering, refers to the force tending to compress a body, and is generally expressed as follows: (1) In pounds per square inch; (2) In pounds per square foot; (3) In inches of mercury; (4) In feet or inches of water.

The values of these different units of pressure are as follows:

TABLE SHOWING RELATION BETWEEN PRESSURE EXPRESSED IN POUNDS, AND THAT EXPRESSED IN INCHES OF MERCURY, OR FEET OF WATER.

Pressure in Pounds per Sq Inch.	Pressure in Pounds per Sq. Foot.	70° Fahr.		
		Inches of Mercury.	Feet of Water.	Inches of Water.
1	144	2.0378	2.307	27.68
2	288	4.0756	4.614	55.36
3	432	6.1134	6.921	83.04
4	576	8.0512	9.23	110.72
5	720	10.1890	11.54	138.40
6	864	12.2268	13.85	166.08
7	1008	14.2646	16.15	193.76
8	1152	16.3024	18.46	221.44
9	1296	18.3402	20.76	249.12
10	1440	20.3781	23.07	276.80

The *barometer pressure* is that of the atmosphere in inches of mercury reckoned from a vacuum. At the sea-level, latitude of Paris, the normal reading of the barometer is 29.922 inches of mercury at 32° F. corresponding to a pressure of 14.7 pounds per square inch.

Gauge or Manometer pressure is reckoned from the atmospheric pressure.

Absolute pressure is measured from a vacuum, and is equal to the sum of gauge-pressure and barometer readings expressed in the same units. Absolute pressure is always meant unless otherwise specified.

Pressure below the atmosphere is usually reckoned in inches of mercury from the atmospheric pressure, so that 29.92 inches would correspond to a perfect vacuum at sea-level, latitude 49°.

86. Measurement of Pressure. — Pressure is always measured as a difference, that is, as pressure above or below some other pressure. Although there are apparent exceptions, no instrument has really been devised which actually measures absolute pressure. It would seem at first sight as though the barometer measured the absolute pressure of the atmosphere and thus formed an exception to this rule. What it really does is to measure the difference between the atmospheric pressure and the pressure of mercury vapor and any other gas present in the upper part of the barometer tube. The correction thus made necessary is, however, always negligible.

The common pressure measurements are all made from the atmospheric pressure as a base, that is, they are pressures in pounds, or feet or inches of water or mercury above or below atmosphere. For a knowledge of absolute pressure it is therefore necessary to add to or subtract from the atmospheric pressure at the time of measurement.

The instruments for measuring pressures divide themselves naturally into two classes, those for low pressures, say up to about 15 pounds above atmosphere, and those for higher pressures. To a certain extent the types used for measuring the higher pressures are also used below 15 pounds, but they become less accurate as the pressure to be measured decreases. The low-pressure measuring devices generally take the form known as *Manometers;* those for higher pressures are generally some form of gauge.

87. Manometers. — The term *manometer* is frequently applied to any apparatus for the measurement of pressure, although it is the practice of American engineers to use this term only for short columns filled with mercury, water or other liquid and used to measure small pressures. The pressure is measured in the ordinary case either above or below the atmosphere. To convert this into absolute pressure, find the sum of barometer and manometer readings in the first case and the difference between them in the second case. All readings must be in the same units.

The manometers in common use are glass or metal tubes, either U-shape in form, as in Fig. 121, or straight and connected to a cistern of considerable cross-section, as shown in Fig. 122.

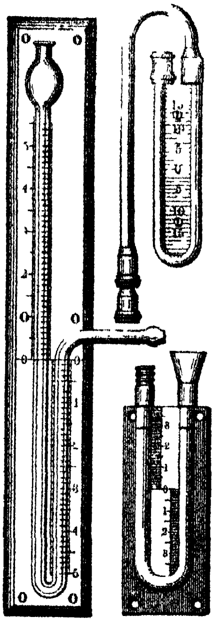

FIG. 121.—TYPE OF ORDINARY MANOMETER.

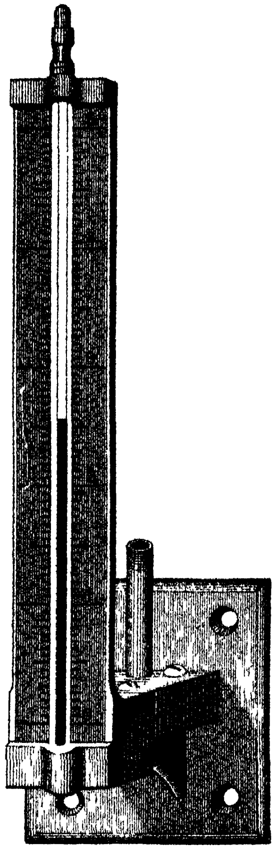

FIG. 122.—CISTERN MANOMETER.

Pressures below the atmosphere can be measured equally well by connecting to the long branch of the tube and leaving the short branch open to the atmosphere.

88. U-shaped Manometer.—In the U-shaped tube, with any form, as shown in Fig. 121, mercury, water, or other liquid is poured in both branches of the tube and the pressure is applied to the top of one of the tubes. When no pressure is applied, the liquid will stand at the same level in both tubes; when pressure is applied, it is depressed in one tube and raised in the other. The pressure above or below atmosphere corresponds to the vertical distance between the surface of the liquid in the two tubes and can be reduced, as explained in Article 85, to pounds pressure per square inch.

An inch of water at a temperature of 70° F. corresponds to a pressure of 0.0361 pound; an inch of mercury, to 0.4905 pound. The principle of action of the **U**-shaped manometer-tubes is as follows: Consider the atmospheric pressure as acting on one side of the tube, and the pressure which is to be measured and which is greater or less than atmospheric as acting on the other side. The total absolute pressure in each branch of the tube must be equal, consequently enough liquid will flow from the side of the greater to the side of the less to maintain equilibrium. Thus let p be the atmospheric pressure and p_1 the absolute pressure to be measured, both expressed in inches of water or mercury; h the height of the column on the side of the atmosphere; h_1 the height on the side of the pressure, the latter both measured above some reference line. Then, if h and p are expressed in the same units,

$$p + h = p_1 + h_1,$$

from which

$$p_1 - p = h - h_1.$$

A very important measurement commonly made by means of **U**-shaped manometers is that of chimney draught. So important is this use that many special types of such manometers have been produced for the purpose. They are generally known as *Draught-gauges.*

A very complete draught-gauge of the **U** shaped manometer type, with attached thermometer and a movable scale the zero of which can be set to correspond to the lower water surface, is shown in Fig. 123 as designed by J. M. Allen of the Hartford Boiler Insurance Co.

A draught-gauge designed by the author is shown in Fig. 124. This gauge is arranged so that one scale will give difference in elevation of the liquid in the two columns. This is accomplished by setting the collar *F* to the lower meniscus of the liquid by the screw *E;*

then by setting the collar *H* to the meniscus of the liquid in the other column by means of the micrometer-screw *R*, the height of the column may be read on the attached scale and the micrometer-screw *R*. The reflection from the two edges of the meniscus enables

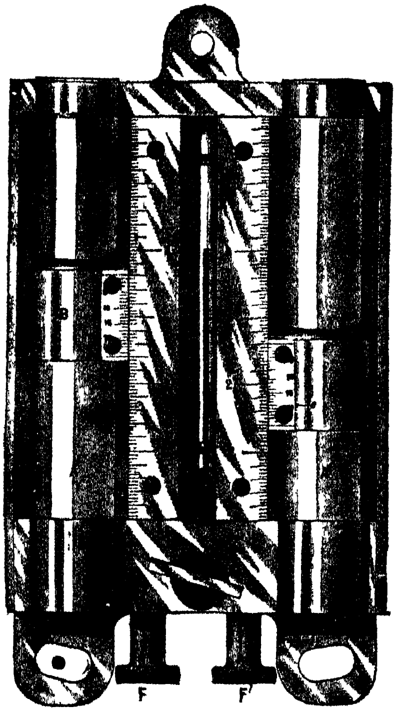

FIG. 123.—DRAUGHT-GAUGE.

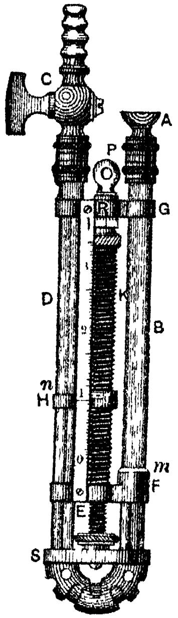

FIG. 124.—DRAUGHT-GAUGE.

the scales to be set with great accuracy. The inches and tenths of inches are read on the attached scale, the hundredths of inches by the graduations of the micrometer-screw *R*.

89. Cistern-manometer.—In the case of a manometer of the form of Fig. 125, the *cistern* or vessel into which the tube is connected has a large area relative to that of the tube. Pressure is applied to the top of the liquid in the cistern, the surface of which will be depressed a small amount, and the liquid in the tube will be raised an amount sufficient to balance this pressure. The pressure above atmosphere corresponds to the vertical distance from the surface of the liquid in the tube to that in the cistern.

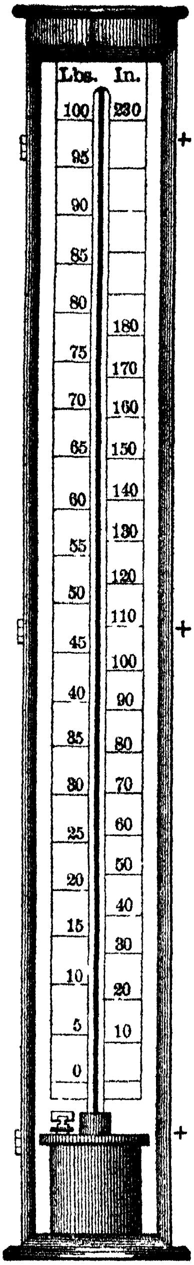

Fig. 125. Mercury Column.

As the liquid is not usually in sight in the cistern, a correction is necessary to the readings in order to find the correct height corresponding to a given pressure. This correction is calculated as follows: Let A equal the area of surface of the liquid in the cistern, a the area of the manometer-tube, H the fall of liquid in the cistern, h the corresponding rise of liquid in the tube, b the height required for one pound of pressure (see Article 85), p the number of pounds of pressure. We have then

$$\frac{H + h}{b} = p;$$

and since the tube is supplied by liquid from the cistern,

$$HA = ha.$$

Eliminating H in the two equations,

$$h = \frac{Apb}{A + a}.$$

If p = one pound,

$$h = \frac{Ab}{A + a},$$

which is the length the graduation should be made to allow for fall of mercury in the cistern and give a value equal to one pound of pressure.

To make this correction applicable the area of cross-section of both tube and cistern should remain uniform.

90. Mercury Columns. — Mercury columns, as used in the laboratories, are usually made on the principle of the cistern-manometer. The tube is very long and made of glass or steel carefully bored out to a uniform diameter. If the tube is of glass, the height of mercury can be readily perceived and read; if of steel, the height of the mercury is usually obtained by a float, which in some instances is connected to a needle which moves around a graduated dial.

In some of these instruments electric connections are broken whenever the mercury passes a certain point, and an automatic register of the reading is made. Fig. 125 shows the usual form of the mercury column, in which the pressure is applied in the upper part of the cistern, so as to come directly on the top of the mercury. In the case of a glass column the graduations are usually made on an attached scale, and are corrected as explained in Article 89 for the fall of mercury in the cistern.

Corrections to the Mercury Column.—The mercury column is usually the ultimate standard by which all pressure-gauges are compared, and its accuracy should be thoroughly established in every particular.

The requirements for an accurate mercury column are:

1. Uniform bore in cistern and tube.
2. Accurate graduations.

The *corrections* to the readings are:

1. *For Inaccuracies of Graduation.*—As it is impossible to make the graduations perfectly accurate, the error in this scale should be carefully determined, and the readings corrected accordingly.

2. *For expansion of the mercury, tube and scale* due to increase of temperature.

The method of correcting for expansion of the mercury and the material enclosing it would be as follows:

Let λ equal the coefficient of lineal expansion of the mercury, and 3λ that of the cubical expansion per degree Fahr.; let δ equal the coefficient of lineal expansion of the metal of the cistern, and δ' that of the material of the tube. Let H' equal the depression in the cistern, h' the corresponding elevation in the tube corresponding to a pressure of one pound, and a difference of level of b'. Let b equal the difference of level corresponding to a pressure of one pound at a temperature of 60° F. Then, as before,

$$h' = \frac{A'b'}{a' + A'} = \frac{A(1 + 2\delta)b(1 + 3\lambda)}{a(1 + 2\delta') + A(1 + 2\delta)} \cdot$$

3. *For the Capillary Action of the Tube.* — This force depresses the mercury in the tube a distance which decreases rapidly as the diameter increases.

The amount of this depression is given in Loomis's Meteorology as follows:

Diameter of Tube Inch	Depression Inch	Diameter of Tube Inch	Depression Inch
0.05	0.205	0.40	0.015
0.10	0.141	0.45	0.012
0.15	0.087	0.50	0.008
0.20	0.058	0.60	0.004
0.25	0.041	0.70	0.0023
0.30	0.029	0.80	0.0012
0.35	0.021		

Variations in the bore of tube might cause slight variations in the value of this correction.

4. There should also be considered a very slight correction, due to the fact that the force of gravity in different latitudes varies somewhat. Since the weight of a given mass of mercury is equal to the product of the mass into the force of gravity, it will vary directly as the force of gravity, or, in other words, the assumed weight of mercury may not be exactly correct.

While it is well to give all these corrections their true weight, yet a false impression should not be incurred concerning their importance. It is hardly probable that the corrections for change in temperature, or corrections for the difference in the force of gravity from that at the sea-level on the equator, would in any event make a sensible difference in the readings of any account in engineering practice.

91. Multiplying and Differential Manometers. — In many cases the ordinary **U**-shaped manometer does not indicate sufficiently small pressure variation for the work in hand. For such purposes it is customary to use multiplying manometers. These divide roughly into two classes: mechanical multiplying manometers and differential multiplying manometers. Descriptions of both types are given below.

As in the case of the U-shaped instruments, many of the best forms of multiplying manometers have been produced for measuring chimney draughts and are known as draught-gauges.

1. *Mechanical Multiplying Manometers.*—A form of multiplying manometer, commonly known as *Peclét's Draught-gauge*, is shown in Fig. 126. It consists of a bottle, *A*, with a mouthpiece near the bottom into which a tube, *EB*, is inserted with any convenient inclination. The upper end of the tube is bent upward, as at *BK*, and connected with a rubber tube, *KC*, leading to the chimney. The tube is fastened to a convenient support, and a level, *D*, is attached. To use the instrument, first level it, note reading of scale, then attach it to the chimney, and take the reading, which will be, if the inclination is one to five, five times the difference of level in the bottle and tube. The scale should be graduated to show differences of level in the bottle, and thus give the pressure directly in inches of water. A commercial form of this gauge is shown in Fig. 127.

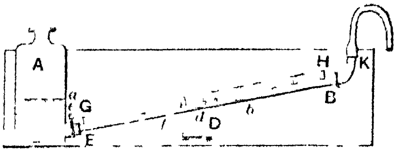

FIG. 126—DRAUGHT-GAUGE.

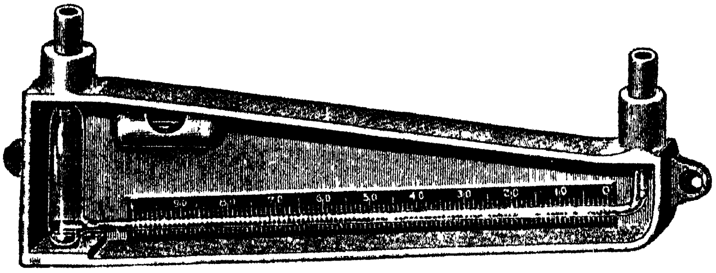

FIG. 127.—MULTIPLYING DRAUGHT-GAUGE

An ingenious modification of this instrument, known as the *Sargent Draft-gauge*, is shown in Fig. 128. The functions of the bottle, *A*, of Fig. 126 are performed by the nickeled brass tube about which the spiral is wound. The spiral which takes the place of the inclined tube of Peclét is made of transparent celluloid and the height of the liquid in it is read by means of the scale carried on the metal reservoir.

Another form of this class of manometers is shown in Fig. 129, as designed by Mr. C. P. Higgins, of Philadelphia. The gauge is filled with water above the level of the horizontal tube, in such a manner as to leave a bubble of air about one-half inch long near one end of the horizontal tube when the water is level in the side tubes. The inside diameter of the vertical tubes being the same, say one-half inch, and that of the horizontal tube one-eighth of an inch, a draught or pressure equivalent to one inch in water, or which will cause the water-level in the vertical tubes to vary one inch, will cause the bubble in the tube to move eight inches in the horizontal tube. In general, the air-bubble moves a distance inversely proportional to the area of the tubes, and hence this gauge can be read more accurately than the ordinary manometer.

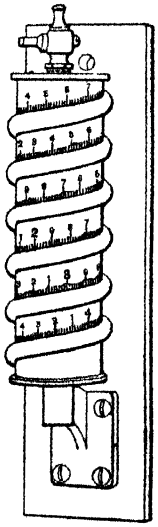

Fig. 128.—Sargent Draught-gauge

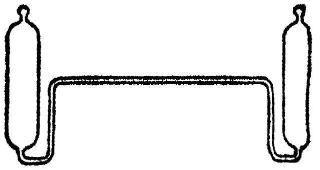

Fig. 129.—Higgins's Draught-gauge.

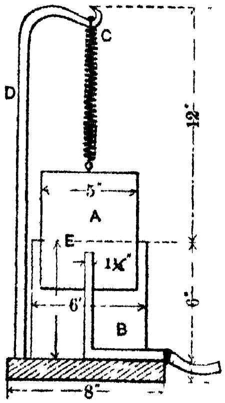

Fig. 130.—Kent's Draught-gauge.

Fig. 130 shows a draught-gauge designed by Prof. Wm. Kent, the dimensions of which are marked on the figure, although they are not material for its operation. The gauge consists of a cup, *B*, which is partly filled with water, and an inverted cup, *A*, suspended above the cup, *B*, by a spring, *C*, with the lower and open end submerged in the water of the cup, *B*. The tube, *E*, extends through the side of the cup, *B*, with its upper end projecting above the surface of the water in the cup, *B*, and is extended by suitable connection to the flue.

By this connection the pressure in the inverted cup, *A*, is reduced to that in the flue where the pressure is to be measured, putting a greater load on the spring, *C*, which causes it to elongate. The amount of elongation will be proportional to the reduction in pressure and can be determined by the use of a suitable scale, the values of which are found by calibration. It is evident that the distance through which the cup, *A*, will move is dependent upon the area of its cross-section and the strength and length of the spring, *C*, and the immersion in the water. Commercial forms of this type of instrument, in which the movement of the cup, *A*, is magnified by various means, are now made.

2. *Differential Multiplying Manometers.* — A class of multiplying manometers in which two liquids of different specific gravities are used are known as *differential manometers*. Their theory can be best approached by first considering a simple but impracticable mechanical multiplying manometer, as shown in Fig. 131. It consists of a **U**-tube enlarged at the upper ends and fitted with the

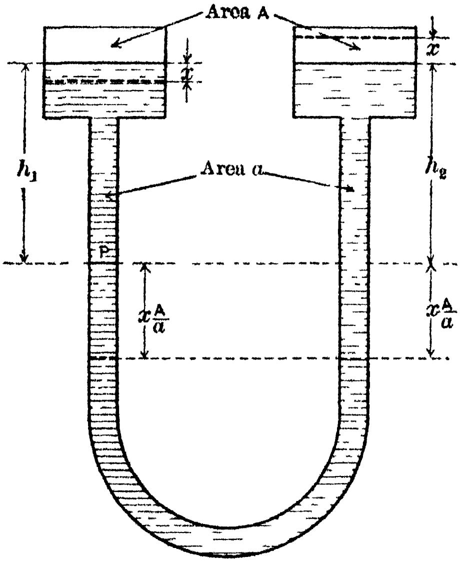

FIG. 131.

small frictionless piston, P, as shown. The device is filled to the level indicated with any liquid, such as water.

If now the same unit pressure p exist above each surface,

$$p + h_1\delta = p + h_2\delta; \quad h_1\delta = h_2\delta,$$

where δ stands for density of liquid.

Increasing the pressure above the left-hand surface by Δp, that surface will sink a distance x and the piston will move down a distance greater in the proportion $\frac{A}{a}$, that is, $x\frac{A}{a}\cdot$

Writing the equation for equilibrium for these conditions

$$p + \Delta p + h_1\delta - x\delta + x\frac{A}{a}\delta = p + h_2\delta + x\delta + x\frac{A}{a}\delta;$$

and since

$$h_1\delta = h_2\delta$$

it follows that

$$\Delta p = 2\,x\delta,$$

which is exactly the same result as for the ordinary U-tube.

If instead of reading the fall or rise of the upper surface, the fall or rise of the piston is read, the actual distance read will be $\frac{A}{a}$ greater than x; *therefore the indication of the instrument can be made larger than that of the ordinary U-shaped manometer for the same pressure difference.*

The use of the solid piston is of course impracticable, so that all actual instruments substitute the film between two non-mixable liquids in its place. In general, however, the two liquids will not have the same densities and the theory of the instrument is then as follows:

To develop the equation for this case, imagine a liquid of density δ_1, to occupy the space above the piston in Fig. 131, and some other liquid with greater density, δ_2, and not mixable with the first, to occupy the rest of the space in the manometer, that is, the right leg and the part of the left leg below the piston. Now remove the piston. With the same external pressure on both legs the surface of contact will move upward or downward, as the case may be, until

$$h_1\delta_1 = h_2\delta_2,$$

the heads h_1 and h_2 being measured above the surface of contact as they were previously measured above the piston.

Applying Δp to the left leg the equation for equilibrium will now be

$$p + \Delta p + h_1\delta_1 - x\delta_1 + x\frac{A}{a}\delta_1 = p + h_2\delta_2 + x\delta_2 + x\frac{A}{a}\delta_2;$$

giving

$$\Delta p = x\left\{\delta_1 + \delta_2 + \frac{A}{a}(\delta_2 - \delta_1)\right\}.$$

In practice the enlargements at the upper ends of the tubes are usually made of metal, so that the movement x cannot be observed. The measurement may be transferred to the dividing surface between the two liquids, provided this surface can be clearly recognized. Where the liquids used are both water-white, as is the case with the pair most commonly used, that is, alcohol and gasoline, it is possible to clearly define this surface and its movement by coloring either one of the liquids.

Since $x\frac{A}{a}$ then becomes the quantity observed, a better form for the equation of the instrument would be

$$\Delta p = x\frac{A}{a}\left\{(\delta_1 + \delta_2)\frac{a}{A} + (\delta_2 - \delta_1)\right\}.$$

The result, since the densities δ are expressed with reference to water, is Δp in water-inches, if $x\frac{A}{a}$ is expressed in inches.

Note that the smaller the factor in brackets, the greater will be $x\frac{A}{a}$ for the same value of Δp; that is, the *smaller* this factor the *more sensitive* will the instrument be. To get a big multiplication, therefore, make $\frac{a}{A}$ as small as possible, and choose two liquids whose densities are very close together.

The actual value of the pressure difference which causes a given movement of the surface of contact may be determined by calculation from the equation above or by actual calibration. The latter method is always preferable.

Another modification of this type is shown in Fig. 132. The heavier liquid, density δ_2, is placed in the lower part of the manometer and the lighter with density δ_1 in the upper part. The two enlargements are connected, as shown, by a tube containing a valve. By opening the valve when the pressure is the same on both upper surfaces the two upper surfaces will assume the same level and the two surfaces of contact will assume the same level. If the valve is then closed and additional pressure Δp applied to the left leg the equation for equilibrium is

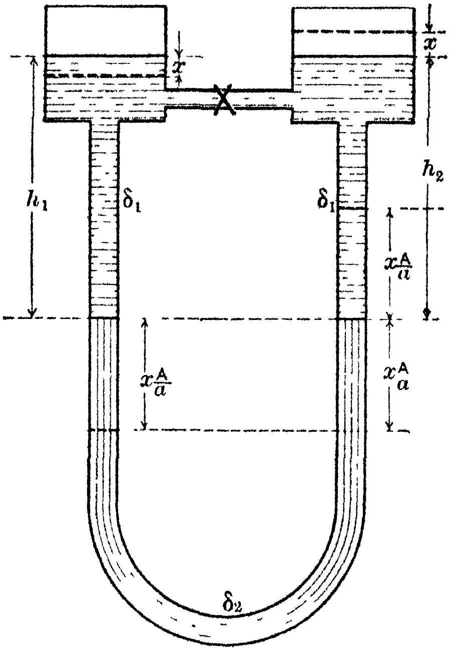

FIG. 132.

$$p + \Delta p + h_1\delta_1 - x\delta_1 + x\frac{A}{a}\delta_1 = p + h_2\delta_1 + x\delta_1 - x\frac{A}{a}\delta_1 + 2x\frac{A}{a}\delta_2$$

and

$$\Delta p = 2x\frac{A}{a}\left\{\delta_1\frac{a}{A} + (\delta_2 - \delta_1)\right\}.$$

In this case $2 \times \frac{A}{a}$ is the total difference in level between the two contact surfaces in the two branches of the tube and is the quantity usually observed. The factors controlling the magnitude of the movement for any given Δp are the same as for the multiplying manometer described just above.

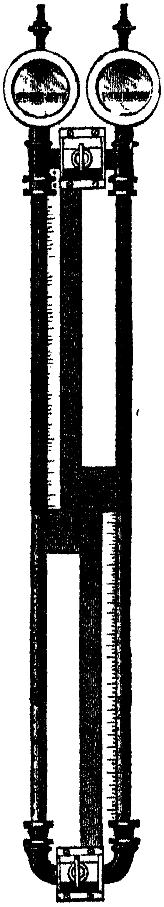

FIG 133. HOADLEY'S DRAUGHT-GAUGE.

The principles of this type of manometer are embodied in the instrument known as *Hoadley's Draught-gauge* and shown in Fig. 133. This gauge was used in the trials of a warm-blast apparatus, described in Vol. VI, Transactions American Society Mechanical Engineers, page 725. It consists of two glass tubes, as shown in Fig. 133, about 30 inches long, and about 0.4 inch inside diameter and 0.7 inch outside, joined at each end by means of stuffing-boxes to suitable brass tube connections, by which they are secured to a backing of wood. The glass tubes can be put in communication with each other at top and bottom by opening a cock in each of the brass connections. Directly over each tube is a brass drum-shaped vessel 4.25 inches in diameter and with heads formed of plate-glass. These drums are connected to the tubes, and also provided with stop-cocks and nipples to which rubber tubes can be attached. Two sliding-scales are arranged along the tubes, one to measure the depression, the other the elevation, of the surface of a liquid filling the lower halves of the tubes. In the use of the instrument two liquids of different densities were used, a mixture of water and alcohol with specific gravity about 0.93 being used for the heavier liquid, and crude olive-oil with a specific gravity of 0.916 for the lighter.

92. Pressure-gauges.—Pressure-gauges in general use are of two classes, known respectively as the *Bourdon* and the *Diaphragm* gauges.

1. *The Bourdon Gauge.* —In the Bourdon gauge the pressure is exerted on the interior of a tube, oval in cross-section, bent to fit

the interior of a circular case; the application of pressure tends to make the cross-section round and thus to straighten the tube. This motion communicated by means of sectors and gears rotates an arbor carrying a needle or hand.

The various forms of levers used for transmitting the motion of the tube to the needle are well shown in the accompanying figures, 134 to 138. The levers are in general adjustable in length so that the rate of motion of the needle with respect to the bent tube can be increased or diminished at will. Thus, in Fig. 134, and also in Fig. 135, the lever carrying the sector is slotted where it is pivoted to the frame; by loosening a set-screw the pivot can be changed in position, thus altering the ratio of motion of hand and spring in different parts of the dial.

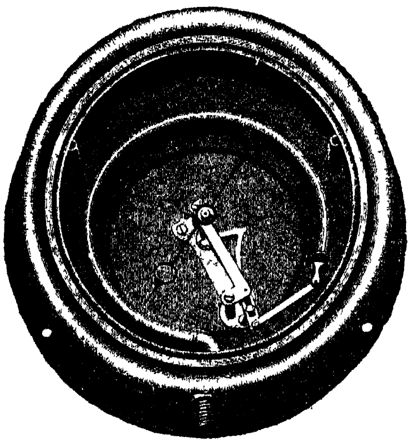

FIG. 134.—BOURDON GAUGE.

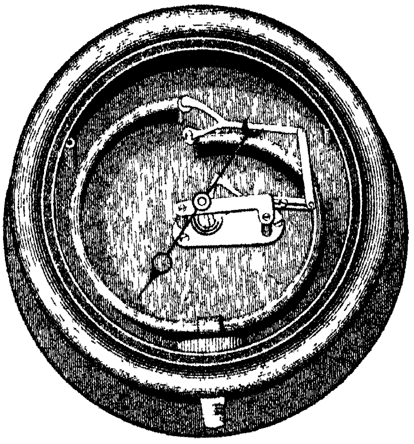

FIG. 135.—BOURDON GAUGE.

Fig. 136 shows a gauge with a steel tube for use with ammoniacal vapors which attack brass. A double tube construction is shown in Fig. 135.

In nearly all these gauges lost motions of the parts are to some extent taken up by a light hair-spring wound around the needle-pivot.

2. *The Diaphragm gauge.*—In the diaphragm-gauge the pressure is resisted by a corrugated plate, which may be placed in a horizontal plane, as in Fig. 137, or in a vertical plane, as in Fig. 138.

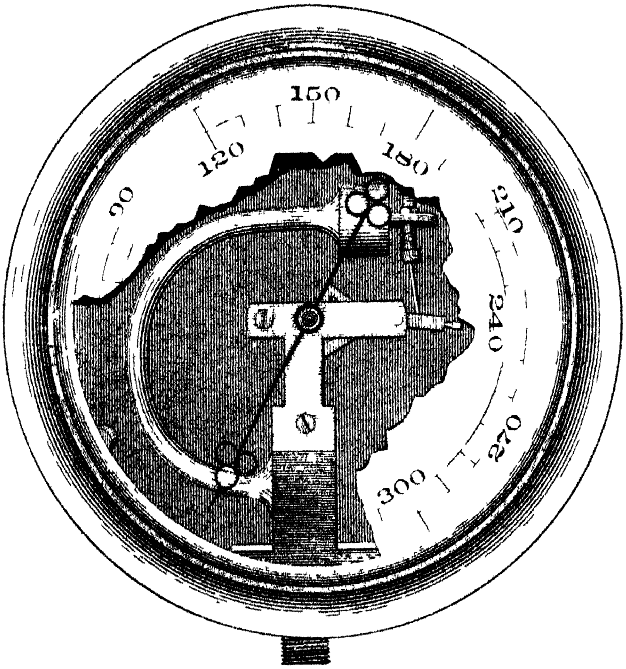

FIG 136 —SCHAEFFER BUDENBERG AMMONIA GAUGE.

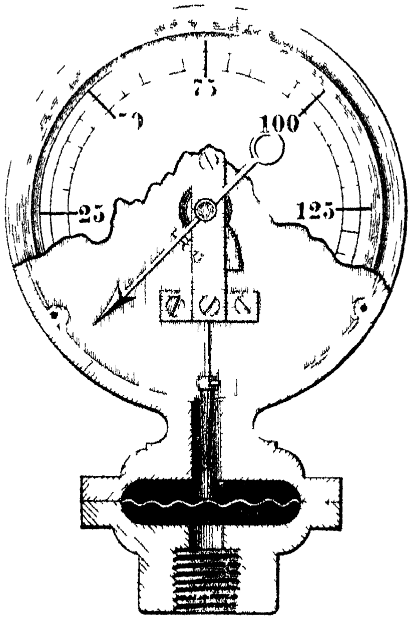

FIG 137 —DIAPHRAGM GAUGE

The motion given the plate is transmitted to the hand in ways similar to those just explained.

In Fig. 137 the pressure is exerted on the corrugated diaphragm below the gauge, and the motion is transmitted to the hand by the rods and gears shown in the engraving.

The construction shown in Fig. 138, in which the diaphragm is vertical, is as follows: the lever is in two parts which are pivoted at the center; one end is fixed to the frame, the other connected to the sector. The center pivot is pressed outward by the action of the diaphragm, drawing the free end downward and rotating the sector, which in turn moves the needle.

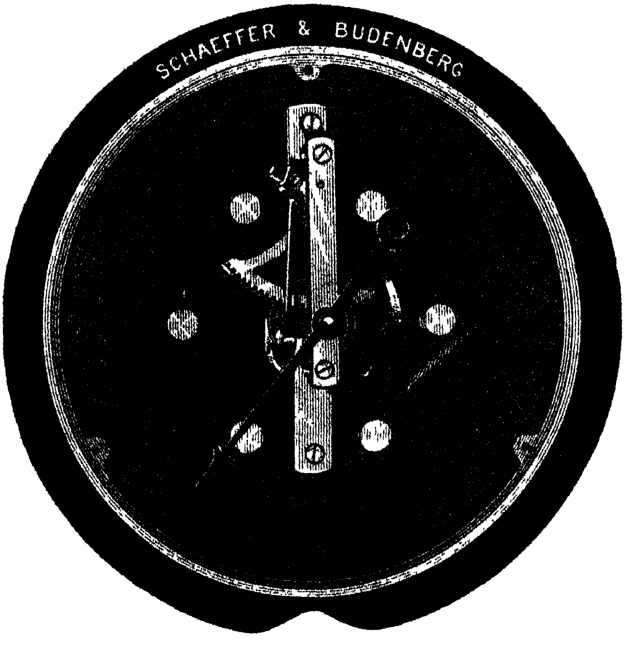

FIG. 138.— DIAPHRAGM GAUGE.

In gauges of usual construction of either class, when there is no pressure on the gauge, the needle rests against a stop, which is placed somewhat in advance of the zero-mark, so that minute pressures are not indicated by the gauge. In the use of the instrument the needle sometimes gets loose on the pivot, or turned to the wrong position with reference to the graduations; in such a case the needle is to be removed entirely, and set when the gauge is subjected to a known pressure. These gauges are also affected by heat. Hence, when set up for use with a heated medium, such as steam, a bent tube or a vessel which will always contain water should be interposed between the gauge and the hot material. All the devices used for thus protecting gauges are termed siphons. The principal types are shown in Fig. 139.

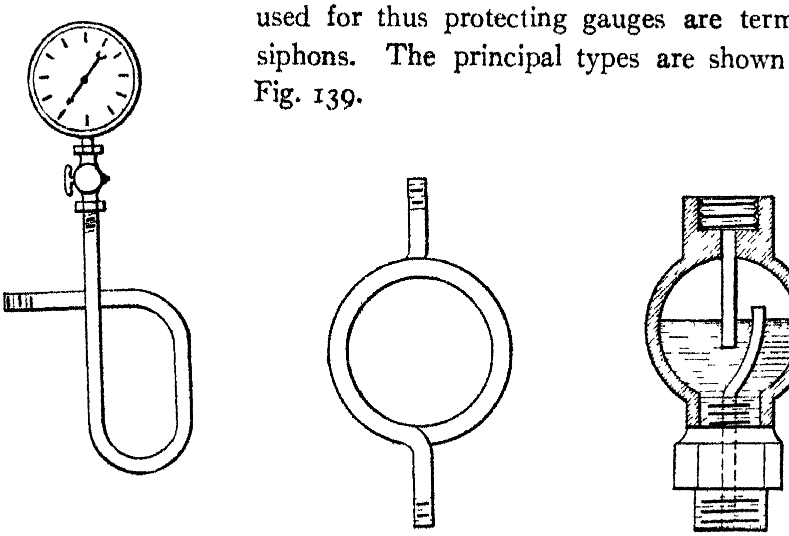

Fig. 139.—Gauge Syphon.

93. **Vacuum-gauges.**—Vacuum-gauges are constructed in the same way as the Bourdon or diaphragm gauges; the removal of pressure from the interior of the bent tube or diaphragm causes a motion which is utilized to move the needle. These gauges are graduated to show pressure below that of the atmosphere corresponding to inches of mercury, zero being at atmospheric pressure. The difference between the reading by such a gauge and that of the barometer, taken at the same time, would be the absolute pressure in inches of mercury.

94. **Recording-gauges.**—Recording-gauges are arranged so that the pressure moves a pencil or pen over a chart which is moved at a uniform rate by clock-work. The Edson recording-gauge is shown in Fig. 140. In this gauge the steam-pressure acts on a diaphragm which operates a series of levers giving motion to a needle moving over a graduated arc showing pressure in pounds; also to a pencil-arm moving parallel to the axis of a revolving drum.

This instrument has an attachment, which is furnished when required, to record fluctuations in the speed of an engine, consisting of a pulley on a vertical axis below the instrument, which pulley is put in motion by a belt from the engine-shaft. On the small pulley-shaft are two governor-balls which change their vertical position with variation in the speed, giving a corresponding movement up or down to a pencil near the lower part of the drum. A diagram is drawn on which uniform speed would be shown by a straight line.

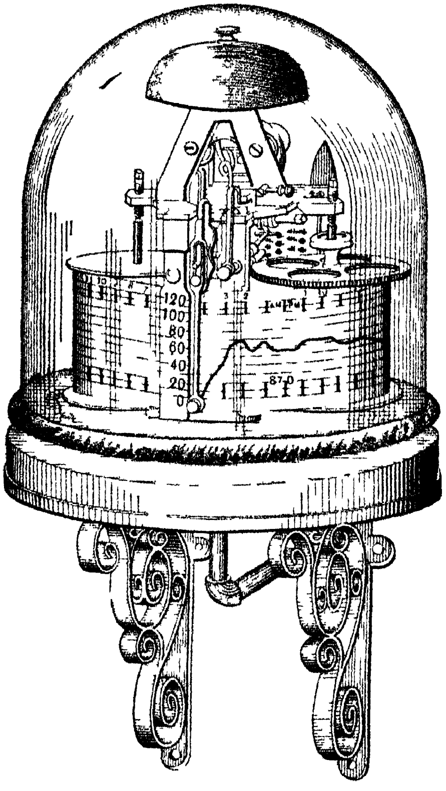

FIG 140 — EDSON RECORDING GAUGE.

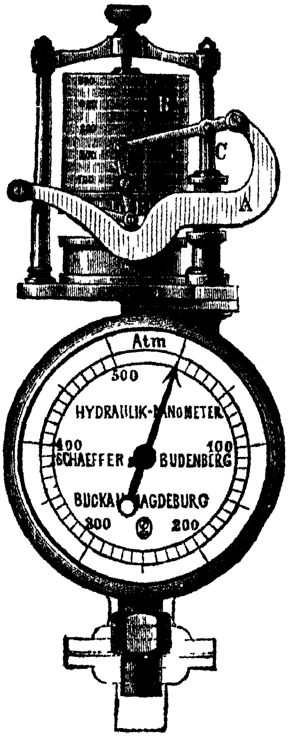

FIG. 141 — RECORDING PRESSURE-GAUGE.

Fig. 141 shows Schaeffer & Budenberg's recording-gauge. This is arranged with a pressure-gauge below the recording mechanism. The drum *B* is operated by clock-work, the piston-rod *C*, which carries the pencil, being moved by the pressure. The pencil-movement is much like that on the Richards steam-engine indicator.

Fig. 142 shows a portion of a diagram made by a recording-gauge. The drum is operated by an eight-day clock, and arranged to rotate once in twenty-four hours. In the diagram the ordinates show pressure, and the abscissæ time in hours and fractions of an hour.

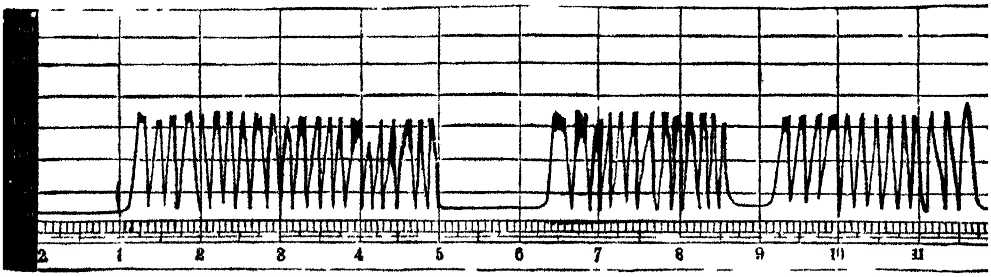

FIG. 142.—DIAGRAM FROM PRESSURE-RECORDING GAUGE.

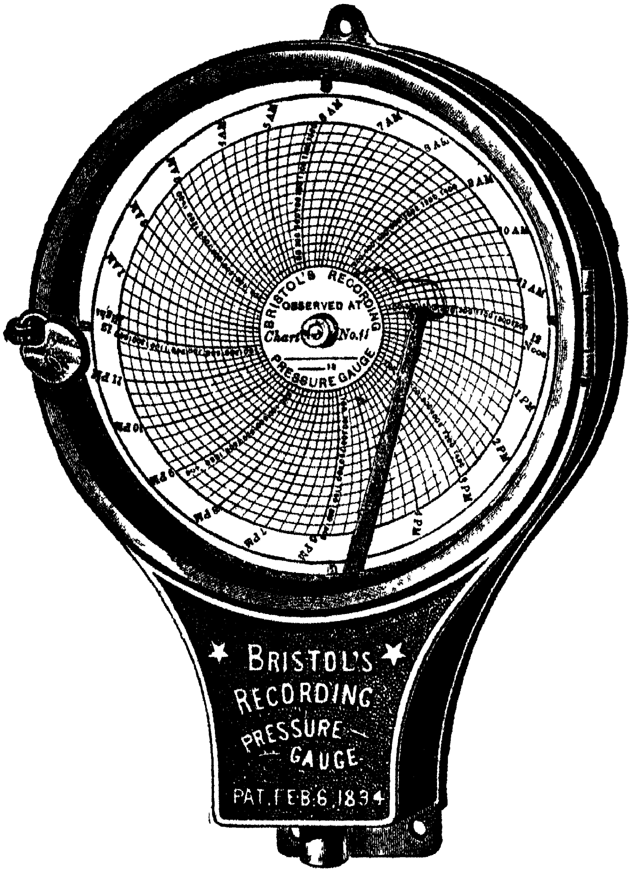

FIG. 143. BRISTOL RECORDING GAUGE.

Figs. 143 and 144 show one type of Bristol recording-gauge. In this instrument the chart is circular and rotates about its center, the pressure ordinates being measured on radiating arcs, as shown.

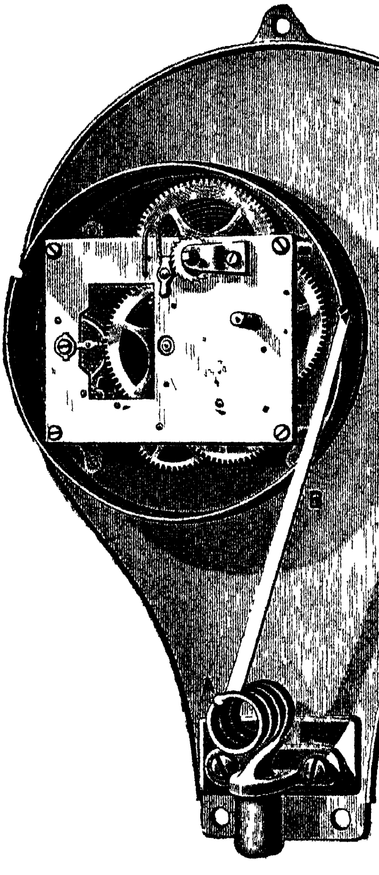

FIG. 144 — BRISTOL RECORDING GAUGE.

95. **Gauge Calibration.** — Gauges are calibrated in two ways: by comparison with other gauges with known error or by comparison with mercury columns or standard weights.

1. *Calibration by Comparison with Other Gauges.* — For this purpose some form of pump is necessary. It generally consists of a cylinder into which a plunger can be driven by rotating a hand-wheel and is fitted with connections for the standard gauge and the gauge to be tested. Fig. 145 shows a portable form of this appa-

ratus. The device shown in Fig. 146 may be used for a similar purpose by putting a standard gauge in place of the manometer shown.

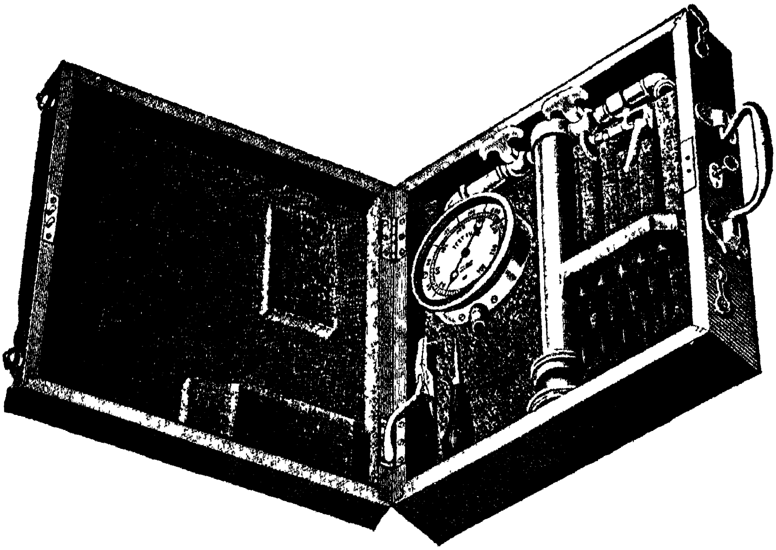

FIG. 145.—PORTABLE GAUGE TESTING APPARATUS.

2. *Comparison with Mercury Column or Standard Weights.*—*Mercury Column.*—The apparatus shown in Fig. 146 may be used for this purpose, connecting the mercury column at E_1. This instrument consists essentially of the cylinder C containing a plunger operated by the hand-wheel D. The cylinder is filled with water or other suitable liquid through the cup shown and then the same pressure applied to both gauge and test column by driving the plunger inward.

Another convenient method is to attach the gauge and the mercury column to a drum in which the pressure can be varied by admitting steam or water under pressure through a throttle-valve.

In all cases the gauge should be tapped before reading and comparison should be made with the standard both with the pressure rising and with the pressure falling. This is necessary to minimize errors due to friction and lost motion in the gauge.

The readings of the mercury column should be corrected in accurate work as outlined in Art. 90.

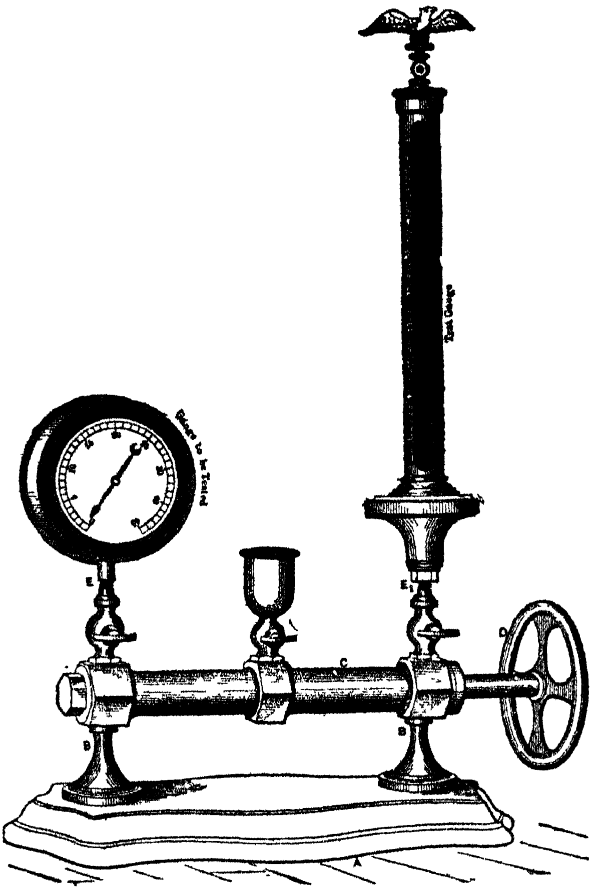

Fig. 146.— Gauge Tester with Mercury Column.

Vacuum gauges are practically always calibrated by comparison with a mercury column. The necessary vacuum may be obtained by means of any suitable air-pump or by connection to a condenser. The U-shaped manometer shown in Fig. 121 is a convenient form for this purpose, but it is necessary that each branch of the tube exceed 30 inches in length.

Comparison with Standard Weights. — There are two forms of this apparatus for this purpose on the market; in one of these the pressure is received on a round piston, and in the other on a surface exactly one square inch in area. The friction in both cases is practically inappreciable; the errors in areas can be determined by comparison with a standard mercury column.

One of the forms with round piston is known as the *Crosby Gauge-testing Apparatus* and is shown in Fig. 147. It is seen to consist of a small cylinder in which works a nicely fitted piston; this cylinder connects with a U-shaped tube ending in a pipe tapped and fitted for attaching a gauge. The tube is filled with glycerine, or oil, in which case a known weight added to the piston produces an equal pressure on the gauge, less the friction of the piston in the tube. This is almost entirely overcome by giving the weight and piston a slight rotary motion.

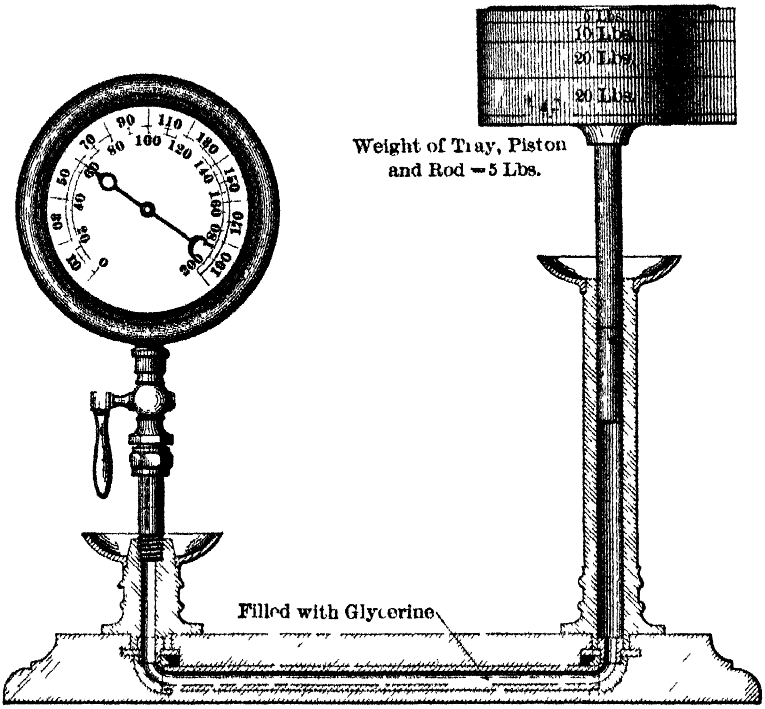

FIG. 147.—STANDARD WEIGHT GAUGE TESTER.

The *Square inch* apparatus consists of a tube the end of which has an area of one square inch enclosed with sharp edges. This tube is connected to the test-pump in place of the standard (see Fig. 146); a given weight is suspended from the center of a smooth plate which rests on the square-inch orifice. The gauge to be tested is connected at *E*, and the pressure applied until the plate is lifted and water escapes from the orifice.

96. Correction of Gauges. — If the calibration shows errors in the gauge, they may generally be corrected; if the error is a constant one, the hand may be removed with a needle-lifter, and moved an amount corresponding to the error, or in some gauges the dial may be rotated. If the error is a gradually increasing or diminishing one, it can be corrected by changing the length of the lever-arm between the spring and the gearing by means of adjustable sleeves or the equivalent. It is to be noted that the pin to stop the motion of the hand is not placed at zero, but in high-pressure gauges is usually set at from three to five pounds pressure.

97. Forms for Calibration of Gauges.

CALIBRATION OF STEAM-GAUGE BY COMPARISON WITH THE MERCURY COLUMN.

Maker and No. of Gauge...

Date...............19 . Observers, { ....

No.	Gauge. lbs.	Mercury Column.				Error. lbs.
		Inches.			Pounds.	
		Up.	Down.	Mean.		
....	.....	....	....	....	....	....
....	....	......	....	......	....	....
......	....	.....	......	.	...	....

Temperature of Room..... deg. Fahr.

Center of Gauge above 0 of column . .ft.

Correction to column reading.. .lbs.

CALIBRATION OF STEAM-GAUGE BY COMPARISON WITH THE SQUARE-INCH GAUGE, OR WITH CROSBY'S GAUGE-TESTING APPARATUS.

Maker and No. of Gauge...

Date...........19 . Observers, { ....

No.	Actual Pressure lbs. per sq. in.	Gauge.	Error.	Remarks.
....	....	....	....	....
....	....	....	....	....
....	....	....	....	....

CHAPTER VII.

MEASUREMENT OF TEMPERATURE.

98. **Temperature.** — The term "temperature" is not easy to define. As a matter of fact no definition strictly accurate has ever been given, but to say that temperature is a measure of the tendency for one body to transmit heat to another is probably open to least objection.*

99. **Thermometers and Thermometric Materials.** — Instruments designed to measure temperature are generally called thermometers, but, depending upon the materials used or the principle of operation, special names are often assigned to the different instruments.

In general, what is really done in measuring temperature is to observe the temperature of the material of which the instrument is made, the construction being such that the thermometer acquires the temperature of the medium in which it is placed.

There are certain properties of matter, like lineal or volume expansion, electrical resistance, or the electromotive force set up in a thermopyle, which vary continuously with the temperature, and any of these may therefore be used to measure temperature.

As a consequence, the number of available *thermometric* materials, as well as the variety of methods by which they may be employed, is quite large. The former comprise gases, liquids and solids, and as far as the methods are concerned the following is a fairly complete list:

1. Expansion of a gas under constant or under variable pressure. This method is very little used.

2. Increase in the pressure of a gas when heated at constant volume. Of this class are all of the well-known gas thermometers, such as the air and the hydrogen thermometer.

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* See Edser, Heat, for Advanced Students.

3. Expansion of a liquid under constant or variable pressure. The ordinary mercury and alcohol or ether thermometers belong to this class.

4. Increase in the pressure of a completely confined liquid, such as mercury. This method is also little employed.

5. A method utilizing the relation existing between the vapor-pressure of a given liquid and its temperature. Instruments built on this principle are known as thalpotasimeters, but are not much used.

6. Expansion of solids. This usually consists in noting the relative expansion of two dissimilar solids with respect to each other. The instruments are usually known as expansion pyrometers and quite widely employed.

7. Fusion methods, which usually consist in exposing a series of materials of known fusing-points to the effect of the temperature to be measured and noting which of the series fuse.

8. Calorimetric methods, in which a body whose temperature is to be measured is mixed with a liquid having a different temperature. The temperature of the body may then be computed by the law of mixtures from the variation in temperature of the liquid.

9. Electric methods, either determining the change in resistance of a given length of conductor as the temperature changes, or measuring the electro-motive force set up when a thermo-couple is exposed to the temperature to be measured. The former are usually known as resistance thermometers, the latter as electric pyrometers. These instruments with proper handling are very accurate and much used.

10. Optical methods. The instruments based upon optical methods may be of two kinds: those measuring *total radiation* from a given area of the hot body, and those which depend for their action upon the intensity of the *luminous radiation* given off. The former are often called *radiation pyrometers*.

It will thus be noted that there is a great variety in the methods which may be used to measure any given temperature. But, depending upon where this temperature is located in the range, that is, low, moderate or high, it will usually be found that one or two methods should have the preference. The most reliable range for each method or instrument will be pointed out in what follows.

It will also be noted that the range of *thermometric materials* is quite wide, comprising, as it does, gas, liquids and solids.

100. Thermometric Scales and Thermometric Standards. — In all methods of measuring temperature, it is usual to establish, by arbitrary selection, two fixed standard temperatures, and then to divide the interval between them into a certain number of parts, called degrees. Temperatures higher or lower than these are then stated in the same *thermometric scale*, by simply continuing the subdivision into degrees above or below the two standard fixed points.

For the great majority of engineering work in this country the scale used is the *Fahrenheit.* In this scale the interval between the two standard fixed temperatures, which are usually the temperature of melting ice and that of water boiling under standard atmospheric pressure, is divided into 180 equal parts or degrees. The temperature of melting ice (the freezing-point) being called 32 degrees in this scale, the boiling-point of water under the conditions stated will then be 212 degrees. In the *Centigrade* scale the freezing-point is called 0 degrees and the boiling-point 100 degrees, there being 100 equal divisions or degrees in this range. The latter scale is much used for scientific work. Finally, there is a third scale, that of Réaumur, in which the freezing is again called 0 degrees, but the interval between the two standard temperatures is divided only into 80 divisions or degrees, so that in this scale the boiling-point of water under standard atmospheric pressure is 80 degrees. This scale is now very little employed.

It is very often necessary to change temperatures from the Fahrenheit into the Centigrade or vice versa. If T_f = temperature on the Fahrenheit scale, and T_c the same temperature as read on the Centigrade scale, then we may write the conversion formula

$$T_f = \tfrac{9}{5}\ T_c + 32. \qquad (1)$$

$$T_c = \tfrac{5}{9}\ (T_f - 32). \qquad (2)$$

Since all of our temperature measurements depend upon the variation of a certain property of the thermometric material with a change of temperature, the choice of a proper thermometric material to be used as a *standard* for comparing the action of all other thermometric materials under the effect of varying temperature becomes very important. It has been found in this connection that hydrogen gas shows a remarkably constant change in the pressure or in the volume, as the case may be, over a very wide range of temperature changes, and this, combined with the fact that the gas is fairly easy to obtain in a pure state, has caused the *hydrogen gas thermometer* to be accepted as the *primary standard.* This instrument, however, is rather delicate and requires a skilled observer, so that it will probably always remain a laboratory standard. For this reason a high-grade mercury thermometer, or other high-grade temperature indicator of less delicate nature than the hydrogen-gas thermometer, is often very carefully compared with the latter, and instruments so calibrated are then often used as standards of comparison, called *secondary standards.* In ordinary laboratory or engineering practice the calibration of thermometers by comparison with secondary standards is by far the more usual case.

101. Gas Thermometers. — One of the simplest forms of gas thermometers is shown in Fig. 148. It consists of a bulb C one end of which is drawn out into a rather fine tube which is usually bent at right angles at F. The rest of the apparatus consists of a manometer FBE and a source A of supply of mercury or other liquid to measure the pressure of the gas volume confined in C. At a there is either a scratch on the glass or, in the better grade of instruments, a black glass tip is fused in the side of the tube, to serve as an indicator for the constant volume of gas. The bulb C for the lower temperature range is usually of glass; for the higher ranges generally of porcelain.

The principle of the constant-volume gas thermometer is very simple. For any two states of a perfect gas we may write

$$\frac{pV}{T} = \frac{p_1V_1}{T_1}, \qquad (3)$$

where p, V and T represent absolute pressure, volume and absolute temperature respectively.

Now in the constant-volume gas thermometer, $V = V_1$ for any two observations, hence

$$\frac{p}{T} = \frac{p_1}{T_1}; \tag{4}$$

and if T_1 is the temperature to be determined, we finally have

$$T_1 = T \frac{p_1}{p}. \tag{5}$$

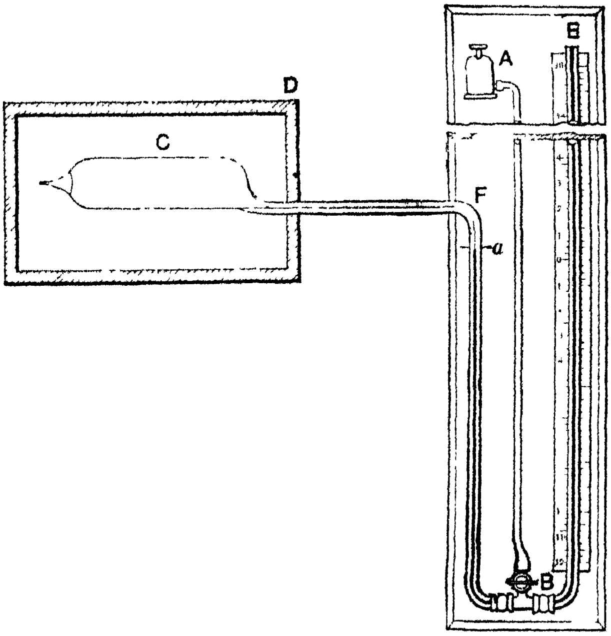

Fig. 148.—Preston Air-Thermometer.

It is therefore necessary merely to observe p for any convenient absolute temperature T and to observe p_1 for the temperature T_1 to be determined. A convenient way to obtain T and p is to use melting ice, in which case $T = 460 + 32 = 492°$.

p and p_1 are absolute pressures, consequently it becomes necessary to take barometric readings when using the instrument. The only reading that is really observed on the instrument is the difference in level of the liquid in the branches of the manometer after the bulb has attained the temperature of the medium to which it is exposed and after the level in the left-hand branch, Fig. 148, has been adjusted to stand at the mark a. Equation (5) may be modified as follows: Let m_1 and m_2 be the height of the liquid levels from any convenient datum line, M, on the side of the bulb. If the pressure exerted by the gas is p and the barometer pressure is b, both expressed in the same units as m_1 and m_2, then we must have

$$p + m_1 = b + m_2, \qquad \text{or } p = b + m_2 - m_1.$$

But $m_2 - m_1 = h$ = difference in level, hence

$$p = b + h.$$

For any other state

$$p_1 = b_1 + h_1,$$

where b_1 may or may not be equal to b.
Therefore

$$\frac{b + h}{b_1 + h_1} = \frac{p}{p_1} = \frac{T}{T_1}$$

or

$$T_1 = T\frac{b_1 + h_1}{b + h}. \tag{6}$$

For the case where melting ice is used to determine T, this reduces to

$$T_1 = \frac{492}{b + h}(b_1 + h_1) = K(b_1 + h_1), \tag{6a}$$

where K is a constant for any given instrument.

Different investigators have used varying forms of the gas thermometer, but the principle is the same in all. The gases commonly employed are air, nitrogen or hydrogen. For reasons already pointed out, the hydrogen-gas thermometer has been accepted as the primary standard temperature-measuring instrument. The range of the hydrogen thermometer is from about $-375°$ F. to $+2700°$ F.

The great objection to the gas thermometer in practice is the great care with which it has to be handled and the fact that temperature determinations with it take considerable time.

The following rules and directions pertain primarily to the air-thermometer, but they hold equally well for any other gas:

Construction of the Air-thermometer. — The bulb of the air-thermometer must be filled with perfectly dry air, as any vapor of water will vitiate the results.

To accomplish this, the bulb is provided with a small opening opposite the capillary tube, which is fused after the dry air is introduced. To effect the introduction of dry air, all the mercury is drawn into the bottle *A*, Fig. 148; the end of the tube *E* is connected to a U-tube about 6 inches long in its branches and about ¾ inch internal diameter, filled with dry lumps of chloride of calcium and surrounded by crushed ice; the opening in the end of the air-chamber is connected by a rubber tube to an aspirator (a small injector supplied with water would act well as an aspirator), and air is drawn through for three or four hours; at the end of this time the bulb and tube should be filled with dry air. While the current of air is still flowing, the cock *B* is opened and mercury allowed to pass into the tubes until it rises to the point *a* in the tube *BF*; the opening in the air-chamber is then hermetically sealed with a blow-pipe, and the connections to the chloride-of-calcium tube removed. This operation fills the bulb with air at atmospheric pressure. By closing the cock *B* before the mercury has risen to the point *a* the pressure will be increased; by closing it after it has passed the point *a* it will be diminished. Packing the bulb *C* in ice, or heating it, will also increase or diminish the pressure as required.

Corrections to Determinations by the Air-thermometer. — The corrections to the air-thermometer are all very small, and affect the results but little if considered. They are:

1. Capillarity, or adhesion of the mercury to the glass. In general the mercury in the two tubes *BF* and *BE* (Fig. 148) is moving in opposite directions, and the effect of adhesion is neutralized.

2. Expansion of the glass. This is a small amount, and may usually be neglected. The coefficient of surface expansion of

glass is 0.00001 per degree F.; it is entirely neutralized if the column of mercury is not reduced in area at the point of meeting the air from the bulb.

3. Expansion of the mercury should in every case be taken into account by reducing all observations to 32° F., the coefficient of expansion being 0.0001 per degree F. Reduce all observations before applying formulæ.

4. Errors in the fixed scale should be determined and observations reduced before applying formulæ.

Directions for Use of the Air-thermometer.

First. To obtain the Constants of the Instruments.—Surround the air-bulb with crushed ice, arranged so that the water will drain off. Note the reading of the mercury column of the air-thermometer h and of the barometer b; by means of the attached thermometers reduce these readings for a temperature of the mercury corresponding to 32° F. Correct for errors of graduation. Divide 492 by the sum of these corrected readings for the constant of the air-thermometer. Call this constant K.

Second. To Measure any Temperature t_1.—Note the corresponding reading of the mercury column h_1, and that of a barometer b_1 in the same room. The reading of the mercury column plus that of the barometer will correspond to $b_1 + h_1$ in the formula

$$T_1 = \frac{492}{b + h}(b_1 + h_1) = K(b_1 + h_1).$$

To obtain from this the actual temperature t_1, subtract 460.

Third. To Compare a Mercurial Thermometer.—Make simultaneous readings of the thermometer, when hanging in the chamber which surrounds the air-bulb, and the height of the mercury column. Perform reduction, and plot a calibration curve for each 10° of graduation.

Fourth. For general use of the air-thermometer, arrange the bulb so that it can be inserted into the medium whose temperature is to be measured with the manometer in an accessible position.

Form for Reducing Air-thermometer Determinations.

TEMPERATURE DETERMINATIONS WITH AIR-THERMOMETER.

By.................................... .... 19......

DETERMINATION OF CONSTANT

	Symbol.	I	II.	III.	IV.
Temperature of air-bulb					
Barometer — Reading					
Thermometer					
Reduced to 32°	b				
Air-thermometer — Reading					
Thermometer					
Reduced to 32°	h				
Constant = $492 \div (b + h)$	K				

DETERMINATION OF TEMPERATURE.

$$t_1 = K(b_1 + h_1) - 460.$$

No.	Barometer.			Air-thermometer			$b_1 + h_1$ Sum.	t_1 Temperature.	Mercury Thermometer.	
	Reading	Ther.	h_1 reduced	Reading	Ther.	h_1 reduced			Reading	Error.
1										
2										
3										
4										
5										
6										
7										
8										
9										
10										
11										
12										

102. Thermometers employing Liquids. — Under this general head may be classed several temperature-measuring instruments of radically different type.

Vapor-pressure Pyrometers or Thalpotasimeters make use of the fact that certain liquids when heated evolve vapors which, if confined, will exert a certain pressure for a definite temperature. Thus in the mercury thalpotasimeter an iron tube is partly filled with mercury which, if heated above a certain point, will give off a vapor which is made to exert a pressure upon a diaphragm, the motion of this diaphragm being communicated to a pointer moving over a scale graduated directly to degrees by calibration, the entire construction being very similar to a diaphragm steam-gauge. Depending upon the range of temperature to which the instrument is to be used, different liquids are used as per following table:

Range, °F.	Liquid.
−85 to +55	Liquid CO_2.
+14 to +212	Liquid SO_2.
95 to 250	Ether.
212 to 440	Water.
420 to 680	Heavy hydrocarbons.
680 to 1200	Mercury.

The main objection to these instruments probably is that they must be exposed full length to the temperature to be measured.

Somewhat similar to the above is the *liquid-pressure pyrometer*, which differs only in that the tube is completely filled with the liquid and strong enough to stand the bursting strain resulting from the tendency to expand. The recording mechanism is the same as in the vapor-pressure instrument. By connecting the reservoir holding the liquid to a very fine capillary tube, the pressure due to expansion may be transmitted some distance (up to 150 feet). In this case varying temperature of the capillary would affect the accuracy of the readings and it would be necessary to apply a correction similar to the stem correction in mercury thermometers. In this instrument it is also necessary to expose the entire length of tube containing the liquid to the action of the temperature to be measured. The mercury-pressure thermometer may be used in the range of − 10 to + 1000° F.

Mercury and Alcohol Thermometers.—Neither one of the two instruments above described are much used, but the third kind under the general head of thermometers using liquids, those employing the *expansion of liquids*, are very extensively used. To this class belong the ordinary *mercury* and *alcohol thermometers*.

The *mercurial thermometer* consists of a bulb of thin glass connected with a capillary glass tube; on the best thermometers the graduations are cut on the tube, and an enameled strip is placed back of them to facilitate the reading. When the mercury is inserted, every trace of air must be removed in order to insure perfect working.* There are certain defects in mercurial thermometers due to permanent change of volume of the glass bulb, with use and time, that result in a change of the zero-point. In a good thermometer the bore of the tube must be perfectly uniform, which fact can be tested by separating a thread of mercury and sliding it from point to point along the tube, and noting by careful measurement whether the thread is of the same length in all portions of the tube: if the readings are the same, the bore is uniform or graduated by trial. In most thermometers the graduations are made with a dividing engine; in some thermometers the principal graduations are obtained by the thread of mercury, as described; in the latter case change in diameter of bore would be compensated. To determine the accuracy of temperature measurements thermometers used should be frequently tested for freezing-point and boiling-point. The accuracy of intermediate points should be determined by comparison with a standard mercurial or air-thermometer.

Mercury freezes at −39° C. (−38.2° F.) and boils at 357° C. (675.6° F.) under atmospheric pressure, so that these temperatures represent the extreme limits of use of the ordinary mercury thermometer. In practice the space above the mercury in a thermometer is very often freed from air in order to prevent oxidation of the mercury. In this case the boiling-point is even lower than 675° F., and the range of use is correspondingly narrower, 550° F. being about the upper practical limit. In order to increase the range, so-called high-reading mercury thermometers are now made of high-grade glass and have the space above the film filled with some neutral gas like nitrogen or carbon dioxide under considerable pressure. The pressure varies, depending upon the range to which the instrument is to be used.* This method of construction of course raises the boiling-point of the mercury and it is thus possible to make thermometers which read up to 1300° F. They are very delicate to handle, however, the glass sometimes breaking from the induced temperature strains alone, to say nothing of the effect of any accidental rough handling.

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* For a detailed discussion concerning the manufacture of mercury thermometers and the errors involved in their use, see Edser, Heat, for Advanced Students

For these reasons 850 or 900° F. is probably about the upper commercial limit of use of such thermometers. Experiments have recently been successfully made to substitute quartz for glass in order to produce a hardier instrument, and mercury-quartz thermometers have been made to read up to 1350° F., but at present they are rather too costly for ordinary use.

The construction of the alcohol thermometer is very similar to that of the mercury thermometer. The expansion of a given volume of alcohol is about ten times that of the same volume of mercury for the same temperature change, and hence an alcohol thermometer is a much more sensitive instrument than a mercury thermometer having the same bulb and tube. Alcohol further possesses the advantage over mercury that it remains liquid down to − 130° C. (−202° F.) and may therefore be used for reading extremely low temperatures. On the other hand, it boils at about 78° C. (172° F.), so that its upper limit is not much above 150° F.

Rules for the Care of Mercurial Thermometers. — The following rules for handling and using mercurial thermometers, if carefully observed, will reduce accidents to a minimum:

1. Keep the thermometer in its case when not in use.
2. Avoid all jars; exercise especial care in placing in thermometer-cups.
3. Do not expose the thermometer to steam heat unless the graduations extend to or beyond 350° F.
4. In measuring heat given off by working-apparatus, or in continuous calorimeters, do not put the thermometers in place until the apparatus is started, and take them out before it

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* See the Bulletin entitled Heat Treatment of High Temperature Mercurial Thermometers, Department of Commerce and Labor, Bureau of Standards, Reprint No. 32, H. C. Dickinson.

is stopped. *Be especially careful that no thermometer is overheated.*

5. In general do not use thermometers in apparatus not fully understood or which is not in good working condition.

6. Never carry a thermometer wrong end up.

7. See that the thermometer-cups are filled with cylinder-oil or mercury. If cylinder-oil is used, keep water out of the cups or an explosion will follow.

8. After a thermometer is placed in a cup, keep it from contact with the metal by the use of waste.

103. Thermometers employing Solids. — *Expansion and Fusion Pyrometers.* — Expansion pyrometers are often called *metallic* pyrometers. The ordinary instruments sold under this name are made of two metals which have different rates of expansion, copper and iron being generally used. The difference in the rate of expansion is employed by means of levers and gears to rotate a needle over a dial graduated to degrees.

In using the metallic pyrometer no reading should be taken until it has had sufficient time to arrive at the temperature of the medium to which it is exposed. When the instrument is first exposed, the needle may be stationary on the dial, or even have a retrograde motion, the heat at first affecting the outside tube only. In order to obtain readings to correspond with the scale on the dial, it is necessary to insert the tube its entire length into the medium whose temperature is to be measured. This is often not convenient and constitutes one of the disadvantages of this type of instrument as usually made.

The *metallic pyrometer* is usually calibrated by immersing in a pipe filled with steam under pressure and comparing the temperature with that given by a calibrated mercurial thermometer. The scale so obtained is assumed to be uniform throughout the range of the pyrometer and beyond the limits of the calibration. Comparison might be made with an air-thermometer. The extreme range of such pyrometers is about 1200° F., but they are probably of little value for temperatures exceeding 1000° F.

Wedgewood's Pyrometer is based on the permanent contraction of clay cylinders due to heating. This contraction is determined by measurement in a metal groove with plane sides inclined towards each other. This pyrometer does not give uniform results.

The type of *fusion pyrometer* is best exemplified by the so-called "Seger Cones." These consist of a graduated series of clay pyramids about 2½ inches high. The composition of the clay differs from number to number in the series. The range of temperature covered is from about 600 to 3400° F., the difference between consecutive numbers being in the neighborhood of 40° F. The method of using these cones consists in exposing a series covering the estimated temperature to the effect of the heat and noting which of them melt down or fuse. Softening on the edges or slight bending over is not to be taken as failure. The last of the series fused is taken to indicate the temperature attained. It will be seen that this method does not give a positive measurement, but it is often used to indicate the temperature of pottery furnaces.

104. Calorimetric Pyrometers. — Pyrometers of this class determine the temperature by heating a metal or other refractory substance to the heat of the medium whose temperature is to be measured. Suddenly dropping the heated body into a large mass of water, the heat given off by the body is equal to that gained by the water; from this operation and the known specific heat of the substance the temperature is computed. Thus, let K equal the specific heat of the body, M its weight; let W equal the weight of water, t its temperature before, and t' after, the body has been immersed; let T equal the temperature of the heated body, t' its final temperature. Then

$$KM(T - t') = W(t' - t).$$

From which

$$T = \frac{W}{MK}(t' - t) + t'.$$

In connection with pyrometrical work, the specific heat of the substance used often has to be determined. The best means of doing this is to heat a piece of the substance of known weight to a known temperature and then to cool it off in a calorimeter as above explained. With proper correction for radiation and water equivalent of calorimeter, *K* may then be computed from the above equation.

The metals best suited for pyrometrical purposes are those with a high melting-point and a uniform and known specific heat. The obvious losses of heat in (1) conveying the heated body to the calorimeter, and (2) radiation of heat from the calorimeter, may be considerable, and should be ascertained by radiation-tests and the proper correction made. Nearly all metals are oxidized, or acted on by the furnace-gases, long before the melting-point is reached; so that, in general, whatever metal is used, it should be protected by a fire-clay or graphite crucible. Platinum, copper and iron are usually employed. The following table gives determinations of melting-points and specific heats:

TABLE OF MELTING-POINTS AND SPECIFIC HEATS OF METALS.

Metal.	Melting-point.		Specific Heat Low Temperatures.
	Degrees Fahr.	Degrees Centigrade.	
Platinum	3110	1710	0.034
Steel	...	...	0.118
Wrought iron	2900	1590	0.110
Cast-iron	2400	1310	0.14
Copper	1980	1083	0.094
Porcelain	.	. .	0.170
Brass	1870	1020	0.094
Zinc	780	419	0.093
Lead	621	327	0.030
Bismuth	507	264	0.030
Tin	450	232	0.047
Mercury	−38	...	0.030
Sulphur	239	115	0.200
Antimony	797	425	...

The mean specific heat of *Platinum* has been the subject of* careful investigation. It was found to vary from 0.03350 at 100° C. to 0.0377 at 1100° C., by Poullet, the experiment being made with a platinum reservoir air-thermometer.

The following table gives some figures for both platinum and copper:

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* See Encyclopædia Britannica, art. Pyrometer.

Platinum		Copper	
Range of Temperature Degrees Centigrade	Mean Specific Heat	Range of Temperature Degrees Centigrade	Mean Specific Heat
0 to 100	0.03350	15 to 100	0.09331
0 " 200	.03392	16 " 172	0.09483
0 " 300	.03434	17 " 247	0.09680
0 " 400	.03476		
0 " 500	.03518		
0 " 600	.03560		
0 " 700	.03602		
0 " 800	.03644		
0 " 900	.03686		
0 " 1000	.03728		
0 " 1100	.03770		

For *wrought-iron* the true specific heat at a temperature t on the Centigrade scale is given as follows by Weinbold.

$$C_t = 0.105907 + 0.00006538\,t + 0.0000000060477\,t^2.$$

Porcelain or *Fire-clay* having a specific heat from 0.17 to 0.[illegible], although not a metal, is well adapted for pyrometrical purposes.

Hoadley Calorimetric Pyrometer.—The Hoadley pyrometer is described in Vol. VI., page 712, Transactions of the American Society of Mechanical Engineers. It consisted of a vessel, Fig. 149, made of several concentric vessels of copper, with water in the inner

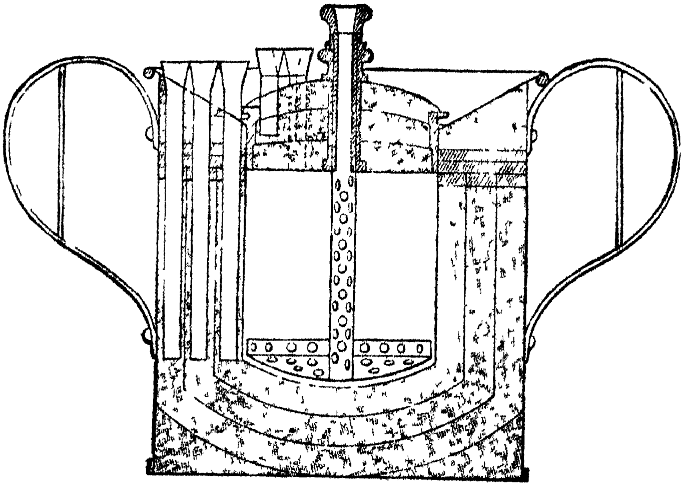

FIG. 149.—HOADLEY PYROMETER.

one, eider-down in the intermediate spaces, and a cover of similar construction. Also a substance to be heated consisting of balls of platinum, or wrought-iron and copper covered with platinum. These balls were heated in a crucible, conveyed to the calorimeter and suddenly dropped in. The calorimeter was provided with an agitator made of hard rubber, with a hole in the center for a thermometer. The balls used as heat-carriers weighed about three-quarters of a pound each; the vessel held about twelve pounds of water.

The balls were heated in crucibles and conveyed to the calorimeter in a fire-clay jar as shown in Fig. 150. The cover of this jar was quickly removed and the balls dropped into the water in the calorimeter.

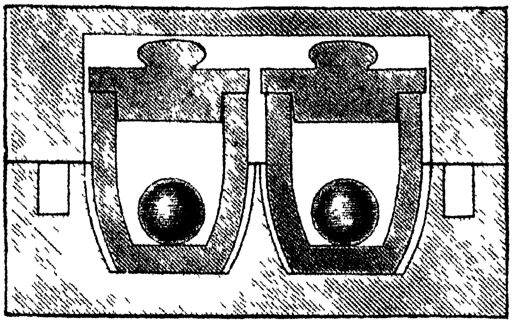

FIG. 150.—PLATINUM BALLS AND CRUCIBLE

With proper handling, calorimeters should give reliable results. There is, however, great danger of error in the transfer of the heated body to the calorimeter, and this, combined with the fact that the indications are slow and require computation, has prevented this method of measuring temperature from becoming general in engineering practice.

105. Electrical Methods of Measuring Temperature.* —There are two classes of electrical pyrometers: resistance thermometers and thermo-couples. The former depend for their action upon the change of resistance of electrical conductors with a change of temperature, the latter upon the fact that the current set up at the junction of a thermo-couple is a function of the temperature to which it is heated. Both methods of temperature measurement are susceptible of great refinement, and of late years several forms of these instruments have appeared which are suited to the requirements of engineering practice.

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* For an extended discussion on Electrical Pyrometers, see High Temperature Measurements by le Chatelier and Boudouard, translated by Burgess.

Resistance Thermometers. — It has been observed that the resistance of electrical conductors generally increases with a rise in temperature. This interrelation may be mathematically expressed so that it is possible to draw a curve between temperature and resistance for any given conductor. It hence becomes necessary merely to determine the resistance in order to read the temperature directly from a curve, but, while this method has been used, commercial instruments are usually so arranged that the scale of the indicating instruments reads directly in degrees. Such a scale may be constructed by direct comparison with a hydrogen-gas thermometer, or by comparison with a secondary standard.

In order to make the instrument reliable it is necessary to use as the conductor a metal which will successfully resist the action of high temperatures and which at any given temperature always has the same resistance. It is found that platinum best meets these conditions, and platinum is therefore widely used.

The best form of platinum-resistance thermometer is probably that of Professor Callendar, who finds that if a coil of pure annealed platinum wire is wound about a mica framework, the whole being protected against the action of gases by an outer tube of hard glass or porcelain (depending upon the temperature to be determined), then the temperature as deduced from an observation of the resistance of the wire will seldom be in error by more than $\frac{1}{100}$° C. at 500° C., while at 1300° C. the error need not be more than $\frac{1}{10}$° C.

Callendar's method of using the resistance thermometer is described by Edser * as follows:

> "The thermometer is supplied with two exactly similar sets of leads, one set T (Fig. 151) being connected to the ends of the spiral of fine platinum wire, while the other set C (called the compensating leads) are joined at their ends within the containing-tube of the thermometer. The Wheatstone-bridge arrangement comprises two equal conjugate arms P and Q. Of the other two arms, one comprises the platinum spiral, the leads T connected across the terminals AB, and

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* Heat, for Advanced Students, p. 401.

the right-hand part of a stretched wire FB; the other arm comprises a set of resistance coils R, the compensating leads C connected across EF, and the left-hand part of the wire FB. Let X represent the resistance of the platinum spiral, R that of the resistance coils, C and T the resistances of the two sets of leads, and $2a$ the resistance of the wire FB, while x is the resistance of that part of this wire between its middle point and the galvanometer connection, so that the two parts of FB have resistances $(a + x)$ and $(a - x)$. Then, since P and Q are equal, the resistances in the other two arms of the bridge must be equal when no galvanometer deflection is produced; in these circumstances

$$R + C + a + x = X + T + a - x.$$

The two sets of leads are exactly similar, being made of the same material and lying side by side, so that their temperatures are always equal, and therefore C is always equal to T. Thus $X = R + 2x$."

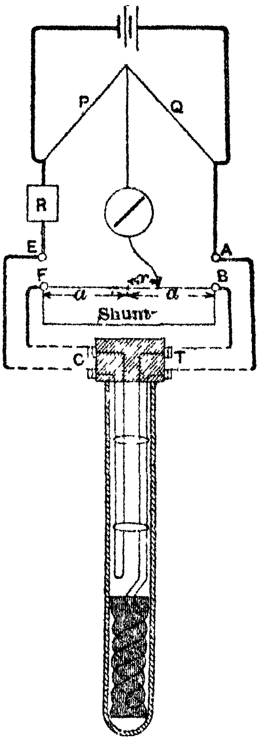

FIG. 151.—PLATINUM RESISTANCE THERMOMETER.

This method determines the resistance of the thermometer in terms of R and X. If, however, the galvanometer circuits were left connected to the middle point of FB, and the resistances so adjusted that with the thermometer cold the galvanometer stands at zero, then with the thermometer at some other temperature the galvanometer would show a deflection, which latter can be expressed directly in degrees, as explained. This is the method ordinarily used in commercial instruments.

In the use of these instruments the following sources of error should be noted:

(*a*) Heating of the Wire by the Measuring Current Itself.—This makes it necessary to keep the current down below a certain limit, as the heating due to the passing current would make the readings too high, or the heating effect must be corrected for. The more sensitive the galvanometer or voltmeter, the smaller need the current be to give a good reading.

(*b*) Lag.—For most work the platinum wire must be protected, usually by porcelain. This makes the wire slow to assume the temperature and causes a *lag*. In general, with the same kind of insulation or protection, the thinner the wire, the less the lag, but of course a very fine wire is apt to fuse more easily. Diameters used are from 0.1 to 0.3 mm. (0.004 to 0.01 inch).

(*c*) Compensation for Resistance of Leads.—In order to prevent the formation of currents at the junction of the leads with the thermometer proper, these junctions must be placed in the cooler part of the circuit. It is usual to employ platinum-lead wires of larger diameter for some ways back than the thermometer wire itself, but even in such a case the varying resistance of these lead wires is apt to cause an error, which is further complicated by the fact that *varying degrees of immersion* will also have an effect. Hence the use of *compensators*, one type of which is shown in the sketch, Fig. 156.

Thermo-couple Pyrometers.—The action of *electric pyrometers* employing *thermo-couples* depends upon the fact that if two strips of dissimilar metals are joined at both ends so as to form a closed circuit, and if one end is heated while the other is kept cool, an electric current is set up which may flow in either direction, depending upon the metals used, the strength of the current depending upon the kind of metals employed and upon the temperature difference between the two ends. We thus have given a direct method of measuring temperature, for, as in the case of the resistance thermometer, a scale for the instrument indicating the current strength may be constructed to read directly in terms of degrees by comparing with any standard temperature indicator.

A variety of metals may be used in the construction of couples, but they are not equally good. The metals should possess a high melting-point and be of uniform composition, and platinum at once suggests itself. As a matter of fact, this is one of the metals used in the high-class *high-resistance type* of instrument, while for the companion metal an alloy of platinum with 10 per cent of rhodium is found to be best. Such couples are usually quite delicate and must be protected by porcelain and iron tubes, as shown in Fig. 152. The rest of the industrial high-resistance electrical pyrometer consists merely of a delicate current indicator, Fig. 153, showing a complete Le Chatelier outfit. The resistance of the outside circuit in an instrument of this type is made purposely high in order to counteract the considerable increase in the resistance of the couple itself as it is being heated. Thus, for instance, the resistance of a platinum platinum rhodium couple, 1 meter long, the wires 0.02 inch diameter, is about 2 ohms when cold, but will increase to 4 ohms at 1800° F. If the outside resistance is 200 ohms and the change in the resistance of the couple is neglected, the error made will be $\frac{2}{200} = 1$ per cent. This will be even less if the couple is not completely immersed. Many industrial instruments, however, have outside resistances of 400 ohms, so that in such a case neither the variation of *resistance in the couple*, varying *depth of immersion*, varying *length of outside leads*, which are usually of much larger diameter than the couple-wires, or a *moderate change in the temperature of the cold junction* can have any serious effect upon the correctness of the indications. On the other hand, because of the small size of couple-wires used and on account of the high total resistance of the circuit, the current set up is very small and the indicating instrument must hence be rather delicate, and must be carefully set up and handled.

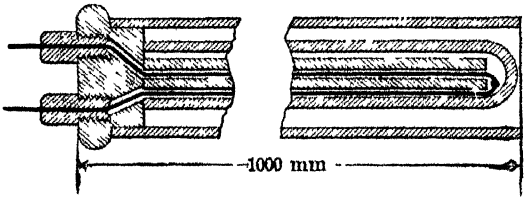

FIG. 152.—ELEMENT OF LE CHATELIER'S PYROMETER

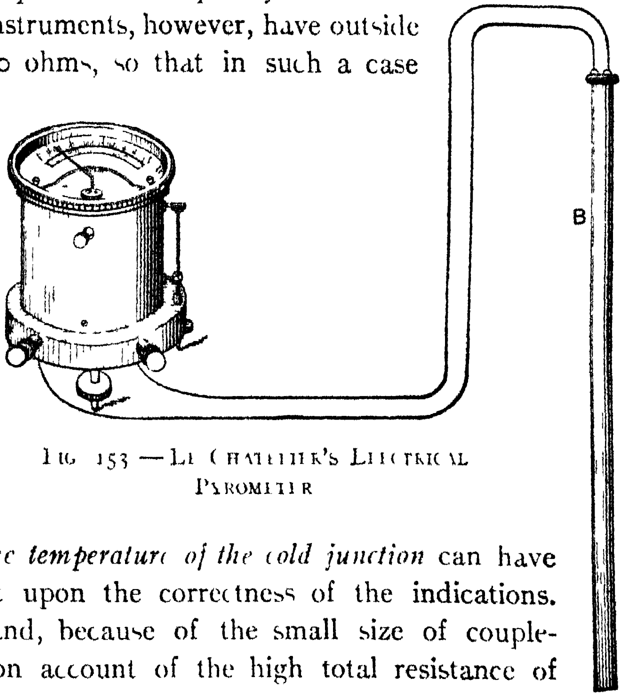

FIG. 153.—LE CHATELIER'S ELECTRICAL PYROMETER

For these reasons successful efforts have lately been made to construct so-called *base-metal* couples which are used in the *low* *resistance* type of electrical pyrometer. Several different makes of such instruments are now on the market, and while they may perhaps not be as accurate as the high-resistance type, they are much hardier and cost quite a little less. The metals used in the couples are usually alloys of tungsten, iron, nickel and copper of composition depending upon the temperatures to be read. The main point in the making of such alloys is to get them of uniform composition, otherwise so-called *parasite currents* will be set up in various parts of the circuit which may render the instrument useless. Since these metals are cheap as compared with platinum, the couples are made of wire of considerable cross-section. This means that the current set up will be comparatively large and that the change in resistance of the couple with change in temperature will be comparatively small. Hence an ordinary low-resistance type of indicating instrument may be used, and the total resistance in such a circuit does not exceed about 10 ohms in most of the industrial pyrometers.

The Bristol type of low-resistance pyrometer with portable instrument is shown in Fig. 154, while Fig. 155 shows the construction of the couples. The latter for protection are in most cases inserted into three-eighth or one-half-inch iron pipe closed at one end. The leads marked in Fig. 154 are of the same metal as the couples, so that the cold junction is really at the points of connection with the indicating instrument. It is possible in most cases to keep the latter near the temperature at which the cold junction was main-

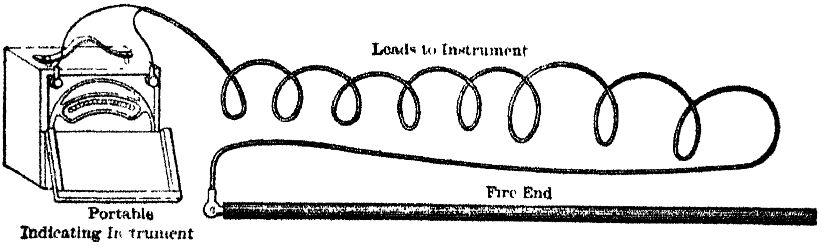

FIG. 154. — BRISTOL PYROMETER

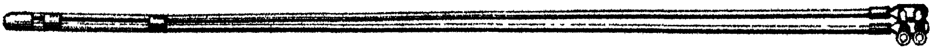

FIG. 155

tained when the instrument was calibrated (usually 75° F.). If this cannot be done, either some correction must be made or *compensators* must be used, for a considerable variation in the temperature of the cold junction causes a much greater error in the low-resistance than in the high-resistance type of instrument. For very accurate work it is of course desirable to take steps to maintain the cold junction at calibration temperature, whatever that may have been, in either type of instrument. If this is not done and compensating devices are not used, it is sufficiently accurate for the low-resistance type to proceed as follows:

If t' = temperature of cold junction of the instrument in use, and t = temperature of cold junction at standardization, then add $t'-t$ to the reading of the instrument if $t'>t$, and subtract $t - t'$ from the reading if $t > t'$. This correction, however, becomes of doubtful value if t' very much exceeds 100° F., if the standardization temperature was in the neighborhood of 75 degrees.

Compensating devices automatically take care of the change of resistance in the low-resistance type of instrument for change of temperature in the outside leads. The device used by the Bristol Co. is shown in Fig. 156. It consists simply of a glass bulb shaped as shown and partly filled with mercury. A platinum-wire loop in series with the outside circuit is fused into the sides of the tube and dips into the mercury. As the temperature in the outside circuit falls, the E.M.F. of the couple would increase on account of the greater range of temperature between hot and cold junction. At the same time, however, the mercury contracts, short-circuiting less of the platinum resistance in the compensator. This increases the resistance in the circuit and thus counteracts the effect of the greater E.M.F. If the temperature of the cold junction increases, the reverse action takes place.

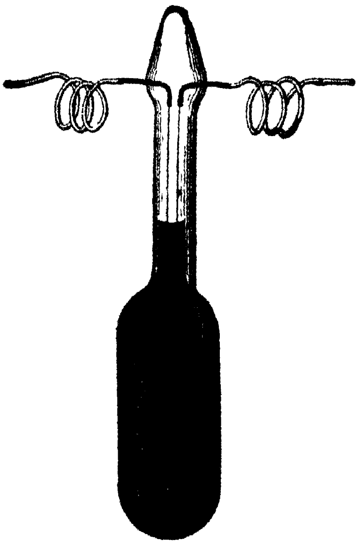

FIG 156. – COMPENSATOR.

106. Optical and Radiation Pyrometers.* — The action of the *optical pyrometer* usually depends upon the comparison of the intensity of the light emitted from the body whose temperature is to be measured with the intensity of the light from a standard source. In order to get rid of difficulties due to color differences and because the laws connecting the temperature of the body with the intensity of the emitted light are simpler, it is usual to deal with only a single wave-length (red is commonly retained). The *Le Chatelier*, the *Wanner* and the *Fery absorption* pyrometers are based on this principle, and the construction is usually such that it is merely necessary to adjust the instrument until two adjacent fields of vision, each illuminated by one of the sources of light, show the same intensity. From the relative displacement of certain parts of the instrument necessary to produce this result, the temperature may be read directly or may be easily computed.

The *Morse thermo-gauge* and the *Holborn pyrometer*, which are very similar in their construction, also belong to this class, although their operation is somewhat different. The former instrument is shown in the conventional sketch, Fig. 157. It employs an incandescent lamp with a rheostat arranged so that the current flowing through it and its consequent brightness may be regulated. The amount of current flowing through is shown by a millivoltmeter connected in circuit, the reading of which can be referred to a scale for the determination of temperature. The lamp is adjusted from an experimental scale for its degree of brightness at different ages.

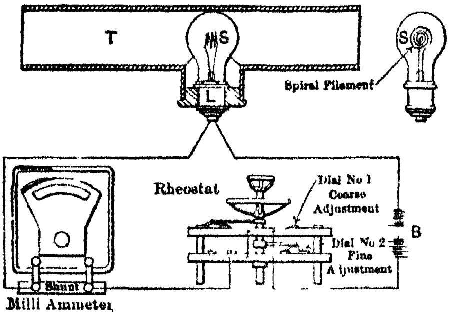

Fig 157.—The Morse Thermo Gauge

In using this instrument the incandescent lamp is located between the eye and the object whose temperature is to be measured, and the current is regulated until the lamp filament becomes invisible.

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* For detailed information concerning optical and radiation pyrometers see le Chatelier's and Boudouard's "High Temperature Measurements."

This instrument is designed for use in tempering furnaces and has an extensive use in that industry.

Still another type of optical pyrometer is the *Mesuré* and *Nouel pyrometric telescope*, Fig. 158, by which the temperature is determined by the direct observation of the light emitted from the hot body.

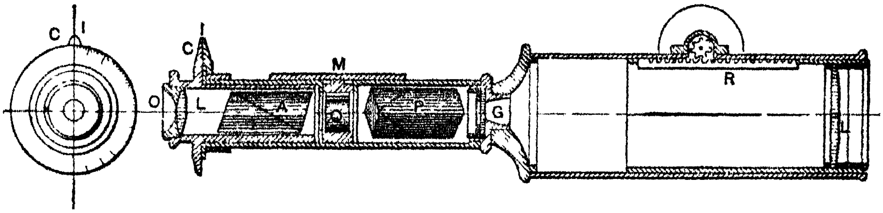

Fig. 158 — Mesuré and Nouel Pyrometric Telescope.

This instrument measures the temperature by taking advantage of the rotation of the plane of polarization of light passing through a quartz plate cut perpendicular to its axis. The angle of rotation is directly proportional to the thickness of the quartz, and approximately inversely proportional to the square of the wave-length.

Light from an incandescent object, passing through the slightly ground diffusing-glass *G* enters a polarizing nicol *P*, and, traversing the quartz plate *Q*, strikes the analyzer *A*, and is seen through the eye-piece *OL*.

In the use of the instrument the analyzer is turned until the object appears to have a lemon-yellow color. The position of the analyzer is indicated by the graduated circle *C*, the reading of which may be referred to a temperature scale. On account of the fact that the change of color from red through lemon yellow to green, or vice versa, which is seen when the analyzer is turned, is not sharply defined, different observers are apt to obtain different results with this instrument and the error may amount to 100° C. For that reason, although the instrument is easy to handle, it should not be depended upon for close results.

In *radiation pyrometers* the energy of total radiation, that of the long invisible rays as well as that of the shorter luminous rays, is measured by the heat effect it produces. The latter may be determined either by thermo-couples, by the expansion of a compound metal strip or by a very delicate resistance thermometer. The

Fery radiation pyrometer, which uses a thermo-couple, may serve as an example of this type, and the following description is taken from the catalogue of the manufacturing company:

"The heat rays given out by a hot body fall on a concave mirror in the pyrometer telescope and are brought to a focus. At this focus is the hot junction of a thermo-couple, and this junction is heated by the focussed heat rays, the hotter the body the hotter the junction.

Referring to Fig. 159, a section of the pyrometer telescope is shown on the right. The mirror *M* receives the heat rays and brings them to a focus at *F*. Here is the hot junction of a little thermo-couple. The cold junction is quite near the hot junction, but is screened from the focussed heat rays. Thus the two junctions are equally affected by changes in air temperature, and the difference in temperature which causes a current to flow will be due to the temperature of the hot body.

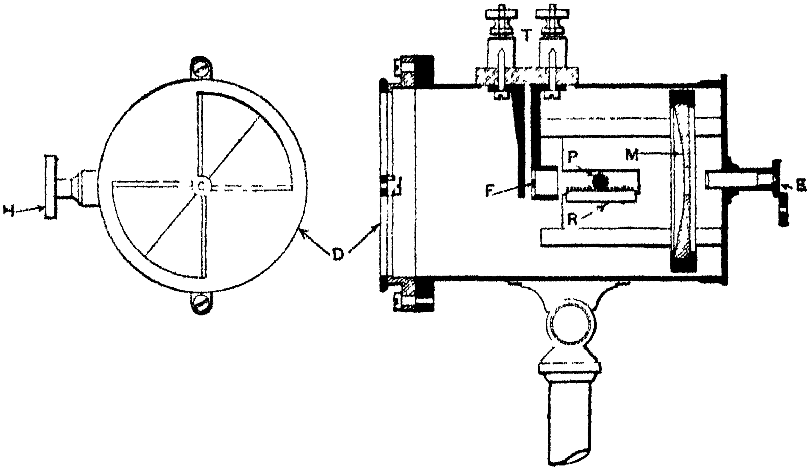

FIG. 159. — FERY RADIATION PYROMETER

The instrument is designed so that within wide limits it is independent of the size of the hot body, or the distance at which it is used.

To guide the pointing of the telescope an eye-piece *E* is provided at the rear end of the telescope through which a reflected image of the hot body can be seen. In the center of the field of view, as seen in the eye-piece, the hot junction of the thermo-couple is seen as a black spot, and this has to be overlapped all around by the image of the hot body. The telescope may be taken as much nearer as is desired without altering the temperature-reading. As the telescope gets nearer to the hot body the mirror *M* receives more heat, but at the same time this greater amount of heat is spread over a larger image, and the intensity of heat remains the same. Thus the only effect of going nearer to the hot body is to increase the amount of overlapping of the image beyond the black spot. The distance can be as much as thirty times the diameter of hot body.

The telescope is focussed by turning the milled head *H* at the side, and this is very simple.

The indicating outfit is provided with an indicator, which is joined to the telescope by a flexible cable. This indicator measures the current generated by the thermo-couple in the telescope, but instead of reading in current it is made to read direct in temperature of the hot body.

For use when the temperature reading reaches the top of the scale, a second scale for higher temperatures is provided, and to use this the diaphragm *D* is swung over the mouth of the telescope.

The stock instrument has two scales, one from 1000° to 2400° F. and the second from 1800° to 3600° F."

107. Calibration of Thermometers and Pyrometers. — The ultimate standard of comparison in the standardization of thermometers and pyrometers is some form of gas thermometer. On account of the fact, however, that such a standard is not easy to handle, secondary standards such as thermometers or pyrometers, which have been compared with the primary standard gas thermometer, are ordinarily used.

The comparison may be made by placing the instrument to be calibrated in a medium which can be slowly heated or cooled, as the case may be, together with the standard. This is probably the usual method for instruments reading to less than 500 or 600° F., as ordinary mercury thermometers. For the latter it is usual also to determine the freezing- and boiling-points of water by methods described below. For high-reading instruments, however, standardizing laboratories, like the Bureau of Standards at Washington, use a high-temperature scale reproduced in terms of certain fixed freezing- and boiling-points of various chemical elements. This temperature scale is not absolutely fixed in the upper ranges, awaiting the results of further investigations, but the provisional scale now used by the Bureau * is as shown in the following table. The scale above 1200° C. is based on the laws of black-body radiation.

		° C.	° F.
Tin	freezing	232	449.5
Zinc	freezing	419	786
Sulphur	boiling	444.7	832.5
Antimony	freezing	630.5	1167
Gold	melting	1064	1947
Copper	freezing	1084	1983
Nickel	melting	1435	2615
Palladium	melting	1546	2815
Platinum	melting	1753	3187

Tin, zinc, lead, antimony and copper must be protected from oxidation, which can be done by melting in graphite crucibles and protecting the surface by powdered graphite. The presence of oxides appears to depress the freezing-points. The freezing-points of lead (327° C.) and of aluminum (658° C. for 99.7 per cent purity) as well as the boiling-point of naphthalene (218° C.) are also often used in standardizing pyrometers.

Calibration of Mercury Thermometers.

1. To Test for Boiling-point. - Suspend the thermometer so that it will be entirely surrounded up to or beyond the reading point in the vapor of boiling water at atmospheric pressure but will not be in contact with the water. Note the reading. From the barometer-reading calculate the boiling-point for the same time. The difference will be the error in position of the boiling-point.

The engraving (Fig. 160) shows an instrument for determining the boiling-point. The bulb of the thermometer is exposed to steam at atmospheric pressure, which passes up to the top of the instrument around the tube, and down on the outside, discharging into the air, or it may be returned directly to the cup, thus obviating the need of supplying water. In the form shown, the parts telescope into each other for convenience in carrying, which is entirely unnecessary for laboratory uses.

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* Bureau Circular No. 7, Oct. 1, 1908, Bureau of Standards.

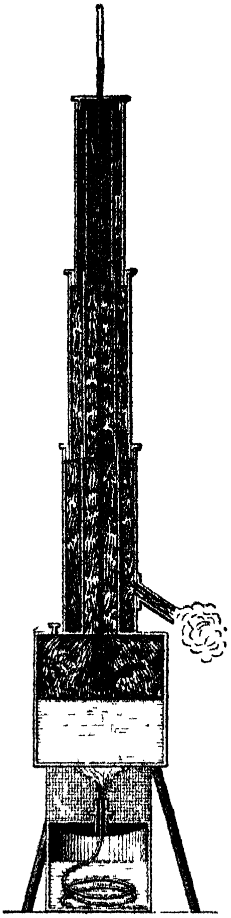

FIG. 160 — BOILING POINT APPARATUS.

2. To Test for Freezing-point. — Surround the thermometer to the reading point by a mixture of water and ice, or water and snow; drain off most of the water. The difference between the reading obtained and the zero as marked on the thermometer (32° for Fahr. scale) is the error in location of freezing-point.

3. For any other temperatures place the thermometer in a liquid bath, such as high-flash-point oil from which all traces of water have been previously removed by heating it, along with a standard which may be a standardized mercury thermometer, resistance thermometer or thermo-couple. Obtain simultaneous readings while heating or cooling slowly. Great care must be taken to thoroughly stir the bath in order to make the temperature uniform throughout.

4. Film or Stem Correction. — The scale on most mercury thermometers is constructed by first locating the freezing- and boiling-points of water under standard atmospheric conditions, and in locating these points both bulb and stem are immersed up to the respective points. In practice, therefore, a thermometer stem should be immersed in the medium whose temperature is being read up to the point where the mercury column in the stem ends. In many cases, however, this is not possible, especially in case thermometer wells or cups are used. In such cases it is always advisable to compute a stem correction because, since the projecting mercury film is at a different temperature than the mercury in the rest of the bulb and stem, the thermometer will be in error by a certain

amount. In many cases such a correction is negligible, but in others it is not, and it is in any important case not safe to depend upon guess work. The correction may be computed from the following equation:

Stem correction $= .00016\ n(T - t)$ for C scale.

Stem correction $= .000088\ n\ (T - t)$ for F scale.

Where n = number of degrees projecting,

T = temperature of bulb,

t = mean temperature of emergent stem.

t is usually found by hanging a small auxiliary thermometer about halfway up the emergent stem; T, however, is unknown, and it is usual to substitute for T the temperature as read from the thermometer. The error thus made is small. It is possible to take care of the stem correction automatically when the thermometer is subsequently calibrated by immersing it in the bath to the depth to which it was used on the test, while the standard is handled in the proper way. This method, however, assumes that the air temperature surrounding the stem is the same during calibration as during use.

Calibration of Pyrometers. — The general method has already been outlined above. The practice of the Bureau of Standards is given in the following extracts from the Bulletin already cited:

Thermo couples, Calibration and Precautions in Use.

(*a*) *Homogeneity.* — For work of precision it is important that each wire of a thermo-couple be of exactly the same chemical composition and physical properties throughout, otherwise the indications of the thermo-couple will vary with the depth of immersion in the heated (or cooled) region. When requested, a special test of the homogeneity of the wires will be made, the results being expressed in terms of the electro-motive forces generated when successive short lengths of the wire are exposed to a constant high temperature.

(*b*) *Annealing.* — Before calibration and use all high-temperature thermo-couples should be annealed by heating them throughout their length, preferably by means of an electric current, to a temperature higher than any to which they may subsequently be exposed. This eliminates the electro-motive forces developed between the hard and soft portions of the wires. When platinum couples have been contaminated by long-continued use they may often be restored to their original condition by annealing (for an hour or more) at high temperatures (1500° or 1600° C.).

(*c*) *Precautions in Use* — The wires of the couple should be fused together at the hot junction, not tied or twisted, as such connections are liable to develop high resistance or to interrupt the circuit when the wires become oxidized. In general the wires of the thermopyle should be protected from the action of hot furnace gases, silicon, metallic vapors, etc. The cold junctions should be so placed that their fluctuations of temperature are negligible. The electrical resistance of the pyrometer galvanometer should be so high that the errors resulting from the resistance of the leads and the variation of the resistance of the couple with temperature and depth of immersion may be neglected. In work of the highest precision at high temperatures (above 1100° C.), contamination of the couples due to the evaporation of rhodium and especially of iridium must be carefully avoided.

A great many of the reported failures of thermo-couples to fulfil the practical requirements of technical applications have been traced to neglect of one or the other of these precautions. Re-tests made by this Bureau of platinum platinum-rhodium couples that have been subjected to long and severe usage in the industries have shown that, after annealing the couples, the new calibrations are in practical agreement with the old.

(*d*) *Calibration.* — Thermo-couples are usually calibrated at the Bureau of Standards, after a thorough annealing, by comparison at four or more temperatures with two standard couples, the couples being immersed for about 25 cm of their length in an electric furnace, and the cold junctions being kept at 0° C.

When a couple is to be used with its cold junction at some temperature other than 0° C., the necessary correction will be indicated in the certificate For the usual forms of thermo-couples made of platinum and its alloys, this correction is approximately $+ \frac{1}{2} t$, where t is the Centigrade temperature of the cold junction, and in general this correction lies between $+ \frac{1}{2} t$ and $+ t$ for practically all types of thermo-couples.

(*e*) *Use of a Pyrometer Galvanometer of Low Resistance.* — In many industrial forms of thermo-electric pyrometer, the electrical resistance of the thermo-couple wires and accompanying leads is not negligible in comparison with the resistance of the indicating instrument. When this is the case the galvanometer does not in general indicate the true E.M.F. of the thermo-couple. If R_1 is the resistance of the thermo-couple wires and attached leads, R_2 that of the galvanometer, and E the true E.M.F. of the thermo-couple, then the E.M.F. = E_1, as indicated by the pyrometer galvanometer, will be $E_1 = E \frac{R}{R_1 + R_2}$. E_1 will thus depend also upon the increase in R_1 due to the increase in the resistance of the heated wires and will, therefore, vary with the depth of immersion of the thermo-couple in the heated space.

Electrical-resistance Thermometers

Calibration. — We may define temperature on the scale of the platinum resistance thermometer as given by

$$pt = 100 \frac{R - R_0}{R_{100} - R_0}, \qquad (a)$$

where R is the measured resistance at some unknown temperature t, and R_{100}, R_0, are the resistances at 100° C. and 0° C., respectively. The relation between the platinum temperature pt and the Centigrade temperature t from − 100° C. to 1100° C. is very exactly given by Callendar's equation

$$t - pt = \delta \left(\frac{t}{100} - 1 \right) \frac{t}{100}, \qquad (b)$$

where δ is characteristic of the kind of metal. For pure platinum $\delta = 1.50$, and it is larger for impure platinum.

The calibration of a platinum resistance thermometer, which is to be used in the range − 100° C. to 1100° C., usually consists in measuring its resistance in melting ice (0° C.), in steam (100° C.), and at one other temperature, usually that of the vapor of boiling sulphur (444 7° C.), and computing other temperatures by means of formulæ (*a*) and (*b*). The values of R_0, the fundamental interval $(R_{100} - R_0)$, and of δ will be given in the certificate furnished by the Bureau. The work of many investigators has shown that a platinum resistance thermometer calibrated at these temperatures may be used to reproduce gas-scale temperatures throughout the range − 100° C. to 1100° C with a degree of accuracy equal to that at present attainable in gas thermometry. For example, when such a calibration is extrapolated to the melting-point of gold, it gives a value (1062° C.) which differs from the true value by an amount which is no greater than the present uncertainty (5°) in our knowledge of this temperature.

If a resistance thermometer is to be used at very low temperatures, the boiling-point of liquid oxygen (− 182.5° C.) may be used to advantage as the third calibration temperature, since the value of δ found from the sulphur boiling-point calibration does not hold exactly at these very low temperatures.

Resistance thermometers which are to be used in calorimetric work for measurement of small temperature changes with high precision will be calibrated at 0°, 100° and 32.384°, the transition temperature of sodium sulphate.

When the construction of a platinum resistance thermometer does not permit of calibration by the above method of using three fixed temperatures, the instrument will be compared directly with the standards of the Bureau at several temperatures in an electric furnace. This method is not in general capable of so high precision as the previously described one. This is the procedure followed when a resistance thermometer and its direct-reading temperature indicator are submitted for test as a single instrument.

Calibration of Optical and Radiation Pyrometers. — The radiation emitted by substances depends on the nature of the substance and condition of its surface as well as upon the temperature. The only body whose radiation depends only on its temperature is the "*black body*," which is approximately realized by a uniformly heated enclosure.

If an optical pyrometer has been calibrated in terms of the radiation from a black body, it will not, therefore, in general give the true temperature of the incandescent body under observation, but nevertheless it will define a consistent temperature scale for any one substance, which in very many cases is all that is necessary in the control of an industrial operation. Where the equivalent black-body temperature is not sufficient, true temperatures may be found by applying a suitable correction, the magnitude of which will depend on the emissive power of the body and on its temperature, or by taking the measurements in such a way that the radiation is very approximately black-body radiation. For example, if the problem at hand is the measurement of the temperature of a furnace or hardening bath, then by inserting a closed-end tube, of suitable material, such as magnesia, porcelain, or tungsten steel, of sufficient length so that the end and some distance along the tube is at the temperature of the furnace or bath, the radiation coming out of this tube is a close approximation to black-body radiation, and the optical pyrometer will then give true temperatures. Again, within many furnaces the conditions approximate fairly close to black-body conditions and the temperatures found by the use of an optical or radiation pyrometer will then differ but little from the true temperature. The readings of optical pyrometers and, to a much greater degree, of radiation pyrometers will be influenced by the presence of flames, vapors, and furnace gases.

The temperature scale defined by the several radiation laws is in agreement with the gas scale throughout the widest range of measurable temperatures, and when these laws are extrapolated to the highest attainable temperatures they are still in satisfactory agreement.

An optical pyrometer may be calibrated by sighting either upon a black body or upon another body whose emissive properties are known. It is, however, necessary to determine the calibration temperatures by some auxiliary means, as a thermo couple; or carrying out the calibration at certain known temperatures, such as the fusing point of gold, palladium, and platinum; or, what is usually the more convenient in the case of industrial instruments, comparing the indications of the pyrometer to be tested with that of a standard instrument, both being sighted upon the same source, which may be a clear furnace, or, in the case of the two instruments using the same colored light, a graphite or metal strip mounted in vacuo and heated electrically. This use of an electrically heated strip permits of a very rapid calibration of sufficient accuracy for industrial and many scientific purposes. With a graphite strip such a calibration may be made up to 1900° to 2000° C. With a strip of tungsten such a calibration might be carried several hundred degrees higher.

The calibration formula for these optical pyrometers which are of the photometer type, and in which light of a single color is used, is very simple. The intensity I of a monochromatic light source, approximating a black body, varies with its absolute temperature T ($= t + 273°$ C.), as follows:

$$\log I = a - \frac{b}{T},$$

where a and b are constants. Such a pyrometer may, therefore, be calibrated completely by finding its readings at two temperatures only, if its construction is otherwise mechanically correct. Often with such pyrometers the monochromatic light is obtained by means of colored glasses which are but approximately monochromatic. In this case the calibration should be carried out at several temperatures, the number depending on the glass used and the accuracy sought.

The relation between the current C through the filament of the lamp of the Morse or Holborn-Kurlbaum type of pyrometer and the temperature t of an approximately black-body source is

$$C = a + b + ct^2, \tag{a}$$

where a, b and c are constants, so that measurements at three temperatures, at least, are necessary to calibrate such pyrometers. The pyrometer lamps should be aged before calibration for some twenty hours at a temperature of about 1800° C.

In order to extend the range of any optical pyrometer above 1500° C. absorption glasses, mirrors, diaphragms, or rotating sectors may be used. It is then necessary to determine the absorption coefficient for the screen used, by the measurement of one or more known temperatures with and without the screen interposed.

The indications of total radiation pyrometers, such as the Fery thermo-electric telescope, obey approximately the law

$$E = k(T^4 - T_0^4), \tag{b}$$

where E is the energy received by the instrument at a temperature T_0 from a black-body source whose temperature is T, and k is a constant. For the measurement of high temperatures, T_0^4 may usually be neglected in comparison with T^4. The industrial forms of these instruments usually give readings departing somewhat from equation (b), so that it is necessary to calibrate them empirically at several points over the range for which they are intended to be used. A relatively large area is usually required upon which to sight these pyrometers These instruments may be used with recording galvanometers and thus give a permanent record of temperatures.

CHAPTER VIII.

THE MEASUREMENT OF SPEED.

UNDER this head is to be understood the determination of strokes or revolutions of any mechanism per unit of time, and not the measurement of the speed of moving masses in general. In practically all pump and engine tests such speed determinations are necessary, for determining either delivered capacity, variation with load, or power.

The number of strokes of a moving mechanism can be counted quite accurately without an instrument if the speed is not too great. This is best done by holding a stick in the hand in such a position that it is struck periodically by some moving part, such as the cross-head of an engine.

The instruments in use for speed determinations fall approximately into three classes:

1. Counting devices.
2. Indicating devices, or Tachometers.
3. Chronographs.

108. Counting Devices. – These are all arranged so that a part of the instrument is moved by direct attachment to the moving machine and they record continuously from the time of attachment until the connection is again severed. For this reason they must always be used in conjunction with a clock or watch if the record is to be reduced to strokes or revolutions per unit of time.

1. *Hand Speed-counter.* — One form of this device is shown in Fig. 101. It consists of a counter operated by holding the pointed end of the instrument against the end of the rotating shaft which should have a so-called "center." In using the instrument, the time is noted by a watch at the instant the counting gears are put in operation or are stopped. A stop-watch is very convenient for obtaining the time. The errors to be corrected are principally

those due to slipping of the point on the shaft, and to the slip of the gears in the counting device in putting in and out of operation. The best counters have a stop device to prevent this latter error, and the gears are engaged or disengaged with the point in contact

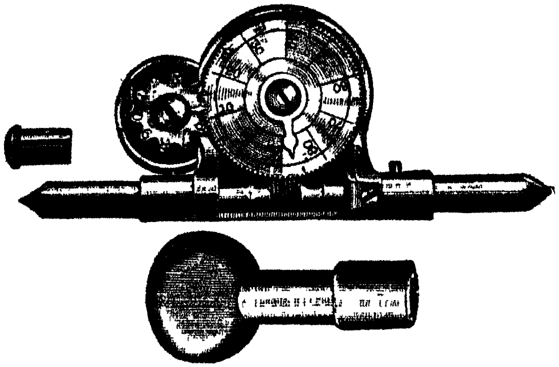

Fig. 161 — Double-ended Hand Speed Counter

with the shaft. To prevent slipping of the point, the end of the instrument is sometimes threaded and screwed into a hole in the end of the shaft.

2. *Continuous Speed-counter.* — The *continuous counter* consists of a series of gears arranged to work a set of dials which show the number of revolutions. The arrangement of gearing in such

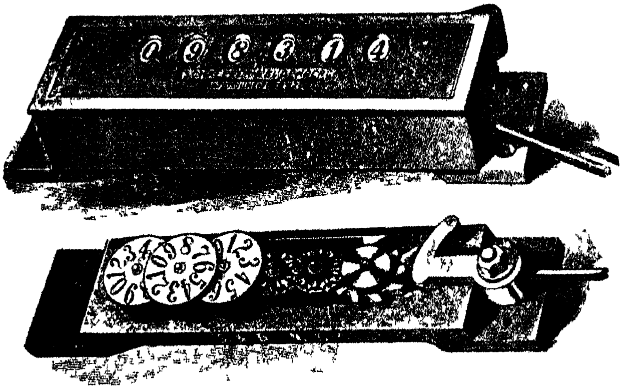

Fig. 162. — Stroke Counter

an instrument is shown in Fig. 162. The instrument can usually be made to register by either rotary or reciprocating motion, and can be had in a square or round case. The reading of the counter is taken at stated intervals and the rate of rotation calculated.

109. Speed Indicators: Tachometers. — All these instruments indicate directly the speed at which the machine to which they are attached is operating. They are thus independent of a time determination and are often more convenient for certain purposes.

1. *Centrifugal Tachometers.* — These instruments utilize the centrifugal force in throwing outward either weights or a liquid. The motion so caused moves a needle or column of liquid a distance proportional to the speed, so that the number of revolutions is read directly from the position of the needle or other indicator.

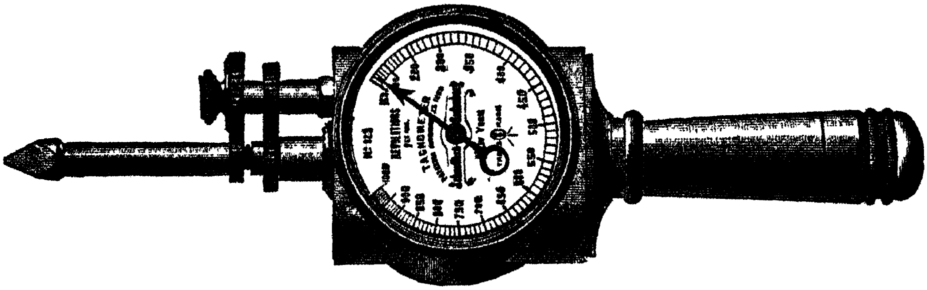

FIG 163 — SCHAEFFER AND BUDENBERG HAND TACHOMETER.

One form of the instrument is shown in Fig. 163. It is arranged with a pointed end to hold against the shaft whose speed is to be determined. A belt driven type is shown in Fig. 164. The movement of the needle over the dial is caused by weights moving under the action of centrifugal force. These instruments are all subject to wear and should always be checked or calibrated for accurate work.

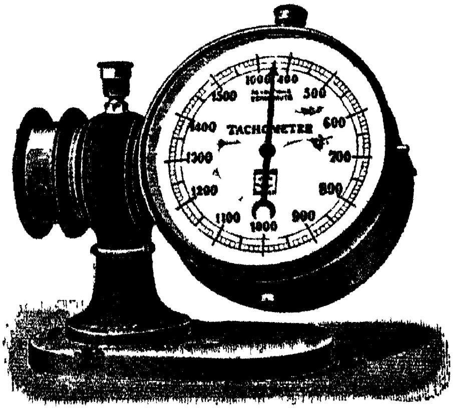

FIG. 164. — BELTED TACHOMETER.

Brown's Speed Indicator consists of a U-shaped tube joined to a straight tube at the center of the bend. The revolution of the U-tube around the center tube induces a centrifugal force which elevates mercury in the revolving arms and depresses it in the center tube. A calibrated scale gives the number of revolutions corresponding to a given depression.

The Veeder Tachometer, Fig. 165, another liquid tachometer, consists essentially of a small centrifugal pump discharging into a vertical tube. The pump is driven from the machine under test and raises the liquid in the tube to an extent proportional to the speed of rotation. The speed is read on a properly graduated scale attached to the tube.

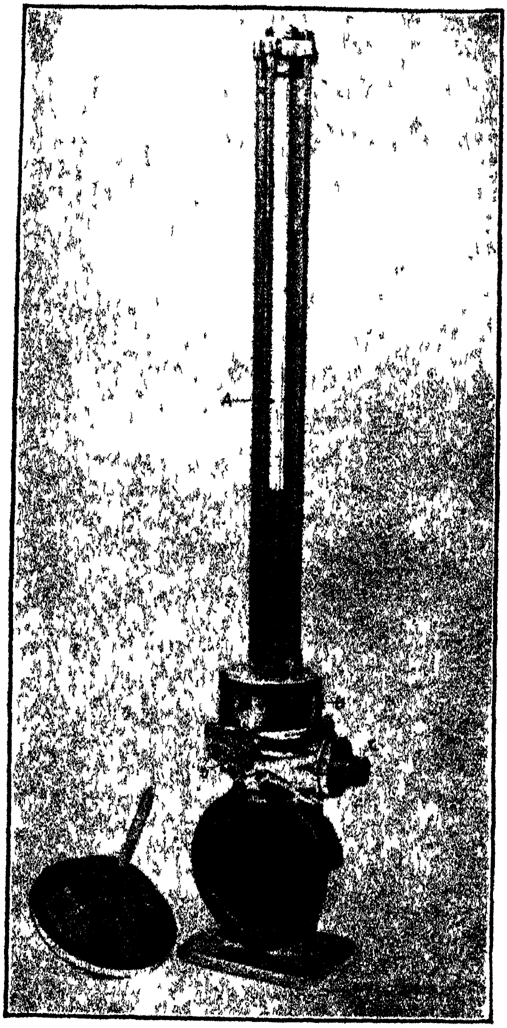

FIG. 165. — VEEDER LIQUID TACHOMETER

Autographic Tachometers. — Variations in speed are shown autographically in several instruments by recording on a strip of paper moved by clock-work the variation in centrifugal force of revolving weights. In the Moscrop speed recorder, shown in Fig. 166, the shaft *B* is connected with the shaft whose speed is to be measured. The variation in the height of the balls near *B*, caused by variation in speed, gives the arm *C* a reciprocating motion, so that an attached pencil makes a diagram, *FED*, on the strip of paper moved by clock-work. The ordinates of this diagram are proportional to the speed.

2. *Frahm's Resonance Tachometer.** — This instrument now made in a great number of forms depends upon the principle of resonance, that is, the property of a body to vibrate in unison with another which has a like period. It consists of a number of steel reeds of different lengths and differently loaded, so that each one of the series has a unique period, mounted in the order of their periodicity, Fig. 167. The variation of period, or of number of vibrations per minute between adjacent reeds, is fixed to suit the purpose for which the instrument is intended.

In use it may rest directly upon the frame of a moving mechanism, it may be connected by belt or mechanical linkage or it may be connected electrically.

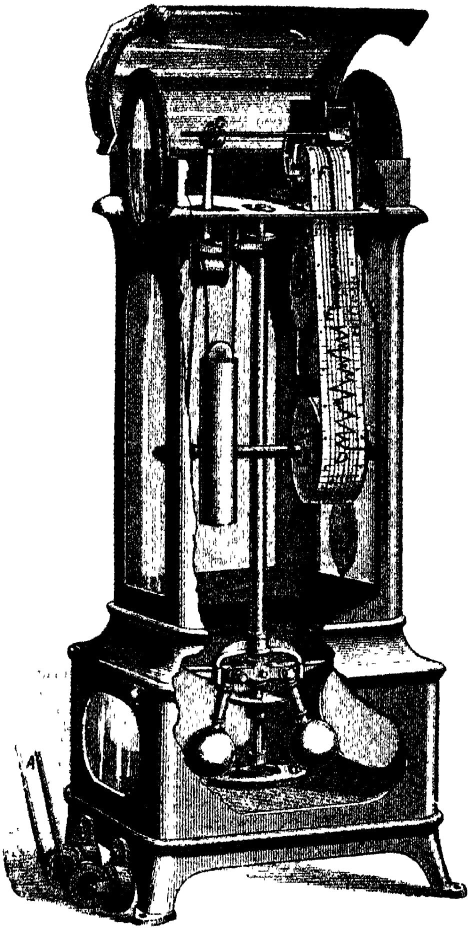

FIG 166. — THE MOSCROP SPEED-RECORDER.

When resting directly on the frame of an engine the impulses due

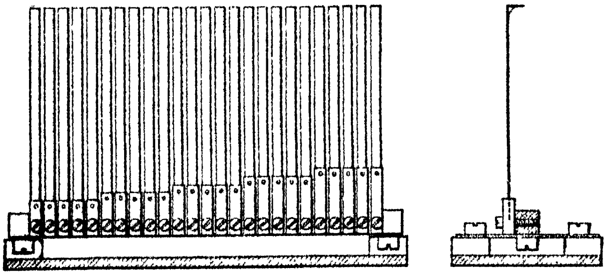

FIG. 167. — REEDS OF FRAHM'S RESONANCE TACHOMETER.

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* See Zeitschrift des Vereins Deutscher Ing. Oct. 15, 1904.

to slight under- or over-balancing are sufficient to set the proper reed in motion.

When belted the shaft of the tachometer carries either a small magnetic or a mechanical device for imparting vibrations to the proper reed.

When connected electrically the reeds are set in vibration by the variation of a magnetic field set up by an alternating current, the alternating current being obtained from a generator driven by or connected to the mechanism of which the speed is to be measured.

110. The Chronograph. — The chronograph,* Fig. 168, consists of a drum revolved by clock-work so as to make a definite number of revolutions per minute. A carriage having one or two pens, *h*, *g*, as may be required, is moved parallel to the axis of the cylinder by a screw which is connected with the chronograph-drum *A* by gearing.

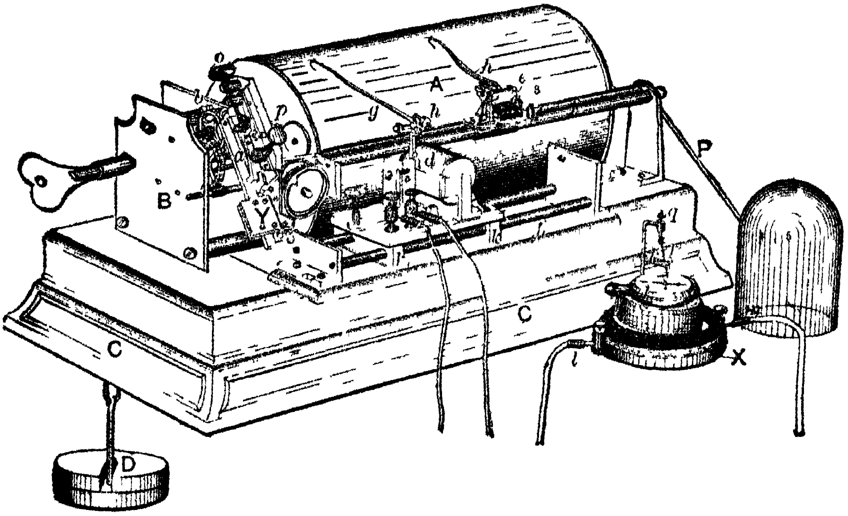

FIG. 168 — CHRONOGRAPH

The pen in its normal condition is in contact with the paper, and it is so connected to an electro-magnet that it is moved axially on the paper whenever the circuit is broken. The circuit may be broken automatically by the motion of a clock, or by hand with a special key, or by any moving mechanism. Two pens are usually employed, one of which registers automatically the beats of a standard clock; the other may be arranged to note each revolution or fraction of a revolution of a revolving shaft. The distance between the marks made by the clock gives the distance corresponding to one second of time; the distance between the marks made by breaking the circuit at other intervals represents the required time which is to be measured on the same scale.

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* See Thurston's Engine and Boiler Trials, page 226.

This instrument has been in use by astronomers for a long time for minute measurements of time, and by its use intervals as short as one one-hundredth (.01) part of a second can be measured accurately.

Tuning-fork Chronograph. — A tuning-fork emitting a musical note makes a constant and known number of vibrations. The number of vibrations of the fork corresponding to the musical tones are as follows:

Note	C	D	E	F	G	A	B	C_2
Vibrations per second..	128	144	160	$170\frac{2}{3}$	192	$213\frac{1}{3}$	240	256

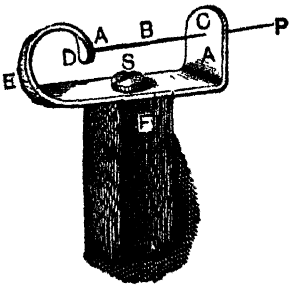

FIG. 169. — STYLUS FOR TUNING-FORK.

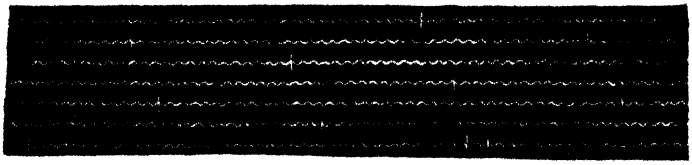

FIG. 170. — SPEED-RECORD FROM CHRONOGRAPH.

If now a small point or stylus be attached to one of the arms of a tuning-fork, as shown in Fig. 169, — in which *F* is one of the arms of the tuning-fork, and *CAED* a piece of elastic metal to which the stylus, *AP*, is attached — and if the fork be put in vibration and the stylus permitted to come in contact with any surface that can be marked, as a smoked and varnished cylinder moved at a uniform rate, the vibrations of the tuning-fork will be recorded on the cylinder by a series of wavy lines, as shown in Fig. 170; the distance between the waves corresponding to known increments of time. If each revolution or portion of a revolution of the shaft whose speed is required be marked on the cylinder, the distance between such marks, measured to the same scale as the wavy lines made by the tuning-fork, would represent the time of revolution.

Fig. 171 (from Thurston's Engine and Boiler Trials) represents the Ranson chronograph; in this case the tuning-fork is moved axially by a carriage operated by gears, and is kept in vibration by an electro-magnet; the operation of the instrument is the same as already described. The form of the record being shown in Fig. 170; the wavy marks being those made by the tuning-forks, those at right angles being made at the end of a revolution of the shaft whose speed is required.

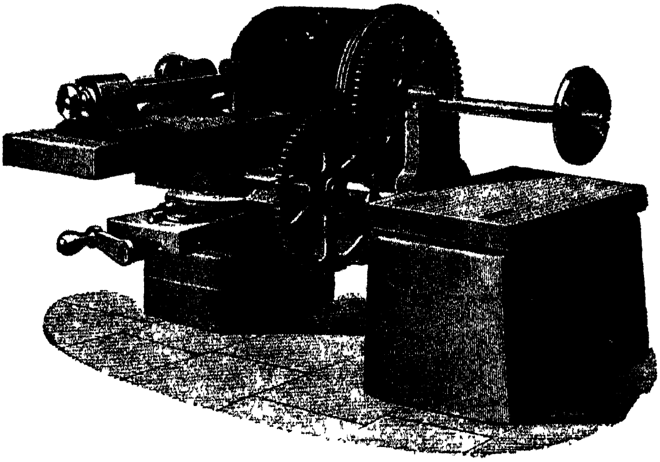

FIG. 171.—TUNING-FORK CHRONOGRAPH

The tuning-fork with stylus attached,* as in Fig. 169, can be made to draw a diagram on a revolving cylinder connected directly to the main shaft of the engine, or the shaft itself may be smoked and afterward varnished. If the fork be moved axially at a perfectly uniform rate, the development of the lines drawn will be for uniform motion, straight and of uniform pitch; but for variations in speed these lines will be curved and at a varying distance apart. From such a diagram the variation in speed during a single revolution can be determined.

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* See Engine and Boiler Trials, page 234.

111. Measurement of Speed Variation within a Revolution. — It is sometimes necessary to measure the variation of engine speed within a revolution. The most convenient method is to use some constant-speed mechanism running at much higher speed than the engine, allowing it to scratch the face of the engine flywheel at equal intervals of time. A tuning-fork has often been used, but is somewhat difficult to handle under these conditions unless arranged so that it is kept in vibration electrically.

In commercial practice an electric motor fitted with a heavy flywheel and driven from a constant-voltage supply, such as a storage battery, has proven satisfactory. It runs at a much higher speed than the engine under test and is arranged to scratch the face of the engine flywheel every time its own flywheel makes one complete revolution. The constancy of angular speed of the engine may then be obtained by measuring the distance between successive marks on the face of its flywheel.

CHAPTER IX.

FRICTION—TESTING OF LUBRICANTS.

112. Friction.—This subject is of great importance to engineers, since in some instances friction causes loss of useful work, and in other instances it is utilized in transmission of power. The subject is intimately connected with that of measurement of power by dynamometers, treated in Chapter X. In connection with these two chapters, the student is advised to read Friction and Lost Work in Machinery and Mill-work, by R. H. Thurston.

Definitions. —*Friction*, denoted by F, is the resistance to motion offered by the surfaces of bodies in contact in a direction parallel to those surfaces.

The *normal force*, denoted by N, is the force acting perpendicular to the surfaces, tending to press them together.

The *coefficient of friction*, f, is the ratio of the friction, F, to the normal force, N; that is, $f = F \div N$.

The *total pressure*, R, is the resultant of the normal pressure, N, and of the friction, F, and its obliquity or inclination to the common perpendicular of the surfaces is the *angle of friction*, whose tangent is the coefficient of friction.

The *angle of repose*, ϕ, is the inclination at which a body would start if resting on an inclined plane. It is easy to show * that for that condition, if W is the weight of the body,

$$W \cos \phi = N; \text{ also, } W \sin \phi = F;$$

and since $f = F \div N$,

$$f = \frac{W \sin \phi}{W \cos \phi} = \tan \phi.$$

See Table 1, Appendix, for values of f and ϕ.

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* See Mechanics, by I. P. Church; p. 164.

It has been shown by experiment that for *sliding friction* (1) the coefficient f is independent of N; (2) it is greater at the instant of starting than after it is in motion; (3) it is independent of the area of rubbing surfaces; (4) it is diminished by lubrication; (5) it is independent of velocity.

113. Classification. — The subject of friction is naturally divided into the following sub-heads, all of which are intimately connected with methods of lubrication:

A. *Friction of rest*, occurring when a body is about to start. It is the resistance to change of position.

B. *Friction of motion*, occurring while two bodies are in relative motion, and being less than the friction of rest.

The second kind, or friction of motion, is of principal importance, and consists of —

1. Sliding friction, occurring usually in one of the following forms:
    - *a*. Bodies sliding on a surface.
    - *b*. Axles or journals revolving in boxes.
    - *c*. Pivots turning on steps.
2. Rolling friction.
    - *a*. One body rolling over a plane.
    - *b*. One body rolling on another not plane.

114. Formulæ and Notation.

α angle of inclination of plane;
ϕ angle of friction;
θ = arc of contact on journal,
β = inclination of force with plane;
N = normal force on a plane;
f = coefficient of friction;
r = radius of journal;
l = length of journal,
n = revolutions per minute.
V = velocity of rubbing in ft. per min.
p = intensity of pressure per sq. in.
P = total pressure;
W = weight of the body.

The most important formulæ relating to friction can be tabulated as follows:

TABLE OF USEFUL FORMULÆ FOR UNIFORM MOTION.

On a Plane.	Force of friction...........	F	$fN = N\tan\alpha = N\tan\phi.$
	Coefficient of friction..... ..	f	$\text{Tan}\,\alpha = \tan\phi = \sqrt{W^2 - N^2} \div N.$
Loose-fitting Journal.	Square of reaction of bearing.	N^2	$W^2 - F^2 = W^2(1 - \sin^2\phi) = W^2\cos^2\phi$
	Weight on journal (squared).	W^2	$N^2 + F^2 = N^2(1+f^2) = F^2\,(1+f^2) \div f^2.$
	Moment of friction	M	$Fr = Wr\sin\phi = fWr \div \sqrt{1+f^2}$
	Work of friction per minute.	U	$FV = 2\pi nrW\sin\phi = 2\pi nrfW \div \sqrt{1+f^2}.$
Perfectly fitting Journal.	Weight on journal (general)..	W	$\int_{\theta}^{+\theta} plr\cos\theta d\theta.$
	Intensity of pressure at $\theta = 90°$	p'	$p \cdot \cos\theta.$
	Weight, perfect fit of journal.	W	$p'lr\int_{-\frac{1}{2}\pi}^{\frac{1}{2}\pi}\cos^2\theta d\theta = 1.57\,p'lr.$
	Pressure per square inch.....	p	$0.64\,W\cos\theta \div lr.$
	Maximum pressure per sq. in.	pm	$0.64\,W \div lr.$
	Total pressure on bearing....	P	$0.64\,W\int_{-\theta}^{\theta}\cos\theta d\theta = 1.27\,W.$
	Total force of friction. .	F	$fP = 1.27\,fW.$
	Moment of friction.. . .	M	$Pfr = Fr = 1.27\,fWr.$
	Work of friction per minute	U	$FV = 1.27\,fWV = 2.54\,\pi fnrW.$

115. Friction of Pivots. —Intensity of pressure $= p$; total pressure $= P$. Moment of friction, $M = \frac{2}{3}fPr$. Work of friction, $U = \frac{4}{3}\pi nfPr$. For a conical pivot, $U = \frac{4}{3}\pi nfPr \div \sin\alpha$. $\alpha = \frac{1}{2}$ angle of cone.

For Friction on a Flat Collar. —Moment of friction, $U = \frac{4}{3}\pi nfP\,(r^3 - r'^3) \div (r^2 - r'^2)$; $r =$ radius of collar; $r' =$ radius of shaft on which it is fitted.

116. Friction of Cords and Belts, Belt Drive. --Suppose, referring to Fig. 172, that a belt passes around a pulley so that T_1 is the tension on the tight side and T_2 the tension on the slack side, the tractive effort, P, of the belt is then $T_1 - T_2$. The frictional resistance to slip is best determined by considering an infinitesimal part δs of the entire arc of contact, S. This small length of arc δs is that subtended by the angle $\delta\theta$ in Fig. 172. This

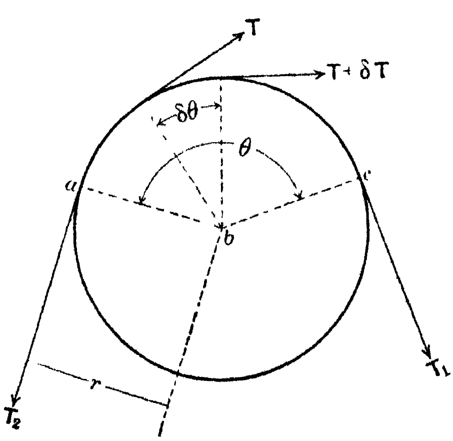

FIG. 172.

arc is reproduced in Fig. 173 for the sake of clearness. The resistance to slipping offered by any two surfaces in contact is the product of the normal pressure, p, on these surfaces multiplied by the coefficient of friction, f. That is, the frictional resistance offered is

$$F = pf.$$

At one end of the arc $db = \delta s$ in Fig. 173, the tension is T, at the other end it may be considered equal to $T + \delta T$, although for our purpose δT may be neglected. The normal pressure, p, is the resultant of the tensions at the ends of this arc δs = arc db. If ad and ab represent these two tensions the resultant, p, is represented by the line ac.

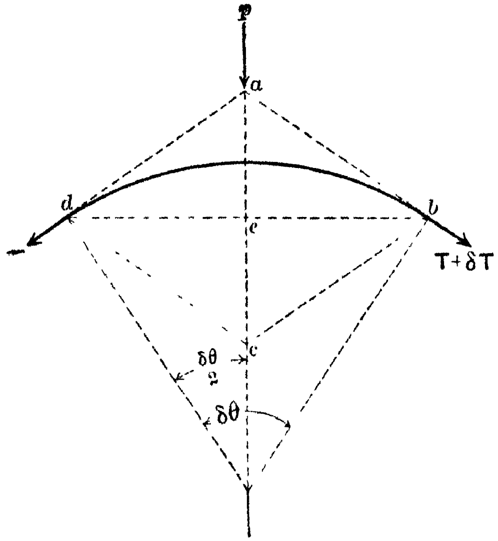

FIG 173.

Now the angle ade = angle $\frac{\delta\theta}{2}$ and $ae = \frac{1}{2}ac = \frac{1}{2}p$,

also
$$\frac{ae}{ad} = \sin\frac{\delta\theta}{2},$$

$$ae = ad\sin\frac{\delta\theta}{2} = T\sin\frac{\delta\theta}{2},$$

from which
$$p = 2\,ae = 2\,T\sin\frac{\delta\theta}{2}.$$

This for our purpose may be written

$$p = T\delta\theta. \qquad (1)$$

The normal force, p, with which the belt presses against the pulley is, however, decreased by the centrifugal force acting on the belt at the same time. The centrifugal force of the infinitesimal arc δs may be expressed by

$$\delta s \, \frac{q}{g} \, \frac{v^2}{r}, \tag{2}$$

where q = the weight of belt one foot long, v = velocity of belt in feet per second, r = radius of pulley in feet, and $g = 32.2$. Now since $\delta s = r\delta\theta$, (2) may be written

$$\text{Centrifugal force} = r\delta\theta \, \frac{q}{g} \, \frac{v^2}{r} = \frac{q}{g} v^2 \delta\theta. \tag{3}$$

The net normal force, p, is therefore

$$\begin{aligned} p &= T\delta\theta - \frac{q}{g} v^2 \delta\theta \\ &= \left(T - \frac{q}{g} v^2\right) \delta\theta. \end{aligned}$$

Multiplying this by the coefficient of friction, f, we have the total frictional resistance

$$F = \left(T - \frac{q}{g} v^2\right) \delta\theta f. \tag{4}$$

On the assumption that any further increase in the tension T causes complete slipping, we must have

$$\delta T = \left(T - \frac{q}{g} v^2\right) \delta\theta f, \tag{5}$$

from which

$$f\delta\theta = \frac{\delta T}{\left(T - \frac{q}{g} v^2\right)}. \tag{6}$$

Integrating (6) between the limits of T_1 and T_2, we have

$$f\delta\theta = \log \frac{T_1 - \frac{q}{g} v^2}{T_2 - \frac{q}{g} v^2}. \tag{7}$$

If the effect of centrifugal force had been neglected, equation (7) would have been

$$f\delta\theta = \log \frac{T_1}{T_2}, \tag{8}$$

in which simpler form the theoretical result is most frequently stated. Equation (8) is safe for moderate speeds where the action of centrifugal force is much less felt.

By putting the tractive force

$$P = T_1 - T_2$$

and without entering into the details of the reduction, the following equations may be derived from equation (7):

$$T_1 = P\left(\frac{e^{f\theta}}{e^{f\theta} - 1}\right) + \frac{q}{g}v^2, \tag{9}$$

$$T_2 = P\left(\frac{1}{e^{f\theta} - 1}\right) + \frac{q}{g}v^2, \tag{10}$$

$$P = \left(T_1 - \frac{q}{g}v^2\right)\frac{e^{f\theta} - 1}{e^{f\theta}}, \tag{11}$$

where e = base of natural logarithms = 2.718.

Again neglecting centrifugal force, equation (11) might have been written

$$P = T_1\left(1 - \frac{1}{e^{f\theta}}\right). \tag{12}$$

117. Friction of Fluids (1) is independent of pressure; (2) proportional to area of surface; (3) proportional to square of velocity for moderate and high speeds and to velocity for low speeds; (4) is independent of the nature of the surfaces; (5) is proportional to the density of the fluid, and is related to viscosity.

Based upon the above laws, the resistance encountered in the case of fluid friction may in general be expressed by

$$S = fA\gamma\frac{V^2}{2g}.$$

in which

f = coefficient of fluid friction (abstract number).
A = area of rubbing surface.
V = velocity of motion of fluid over the surface.
γ = heaviness (density) of fluid.

In this expression, $\frac{V^2}{2g}$ = head producing the velocity V, hence $A\gamma\frac{V^2}{2g}$ = weight of an ideal prism of height h on area A, and this therefore represents a force which if multiplied by f, the coefficient of friction, gives the frictional resistance.

The work of friction per minute,

$$U = SV = fA\gamma\frac{V^3}{2g},$$

if V is expressed in feet per minute.

Viscosity and density of fluids do not affect to any appreciable extent the retardation by friction or the rate of flow, but have some influence upon the total expenditures of energy. Molecular or internal friction also exists.

118. Lubricated Surfaces. — Lubricated surfaces are no doubt to be considered as solid surfaces, wholly or partially separated by a fluid, and the friction will vary, with different conditions, from that of liquid friction to that of sliding friction between solids. Dr. Thurston gives the following laws, applicable to perfect lubrication only:

1. The coefficient of friction varies inversely as the intensity of the pressure, and the resistance is independent of the pressure.

2. The coefficient varies with the square of the speed.

3. The resistance varies directly as the area of journal and bearing.

4. The friction is reduced as temperature rises, and as the viscosity of the lubricant is thus decreased.

Perfect lubrication does not usually exist, and consequently the laws governing the actual cases are likely to be very different from the above. The coefficient of friction in any practical case is likely to be made up of the sum of two components, solid and fluid friction.

Testing of Lubricants.

119. Determinations required. — The following determinations are required in a complete test of lubricants:

1. The detection of adulteration.
2. The detection of acids.
3. The detection of tendency to gum.
4. The measurement of density.
5. The determination of viscosity.
6. The determination of chill point.
7. The determination of flash point.
8. The determination of burning point.
9. The determination of volatility.
10. The measure of the coefficient of friction.
11. The determination of durability and heat-removing power.

120. Adulteration of and Impurities in Oils. — Quantitative determinations of adulteration can be made only by chemical analysis, but there are certain simple tests by which the engineer can sometimes detect adulteration. It should be stated, however, that admixtures are not in all cases to be considered as adulteration.*

Mineral Oils. — The most common case of adulteration in these oils is found in the addition of animal and vegetable oils or mineral soaps to cylinder oils. The presence of animal or vegetable oils in cylinder oils is objectionable because when subjected to the heat and moisture occurring in such service they decompose, yielding fatty acids as one of the decomposition products. Such acids sometimes attack the metal of cylinder and piston and are therefore undesirable. The presence of mineral soaps is objectionable only because they give an erroneous impression of the value of the oil as a lubricant, increasing its viscosity without increasing its lubricating powers.

Adulteration by animal or vegetable oils is detected by mixing with a sample of the oil to be tested an alcoholic solution of sodium hydroxide or potassium hydroxide and then gradually heating to about 200° F. The metal of the hydroxide saponifies the fatty acid of the adulterant and the soap formed separates out.

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* See Friction and Lost Work, by R. H. Thurston.

The presence of mineral soaps can only be detected by incinerating a sample of the oil and judging the purity from the quantity of ash. This should not exceed about 0.1 per cent with a pure mineral oil.

Poorly refined mineral oil may contain sulphur compounds. Their presence may be detected by heating a sample of the oil to a temperature of about 300° F. for fifteen minutes, under which conditions the presence of sulphur is indicated by a considerable darkening of color.

Animal and Vegetable Oils. — Owing to the fact that mineral oils are generally cheaper than those of animal and vegetable origin the former are often used as adulterants for the latter. Their presence can be detected by the saponification method above outlined. In some cases the admixture of mineral oil may be so great that it can be detected by the bloom or sheen which all mineral oil naturally possesses. Under such conditions the saponification test is unnecessary. The detection of bloom is made easy by dropping a small quantity of oil on a "black plate"; when looking at the sample at an angle the iridescent coloring will show strongly if undebloomed oil is present.

Since mineral oil may be debloomed by the addition of nitro-compounds, as nitro-naphthalene or bi-nitro-benzol, the optical test must not be regarded as absolute and in important cases the saponification method must be resorted to.

The common way of detecting the adulteration of animal and vegetable oils among themselves is by using the saponification method and making quantitative determinations. It is necessary for this purpose to accurately determine the quantity of hydroxide necessary to completely saponify the oil and to check with that theoretically necessary for the particular oil under consideration. This is evidently a matter for the chemist and is seldom attempted by the engineer.

Each animal and vegetable oil has a definite specific gravity at a given temperature, and variation from this figure may be taken as an indication that impurities are present. There are several oils, however, whose specific gravity is very nearly the same at the same temperature, and hence this method of identification sometimes fails.

Animal oils may be distinguished from those of vegetable origin by the fact that chlorine gas will turn animal oils brown and vegetable oils white.

121. **Acid Tests.** —Tests for acidity may be made by observing the effects on blue litmus-paper; or better by the following method described by Dr. C. B. Dudley: Have ready (1) a quantity of 95 per cent alcohol, to which a few grains of carbonate of soda have been added, thoroughly shaken and allowed to settle; (2) a small amount of turmeric solution; (3) caustic-potash solution of such strength that 31½ cubic centimeters exactly neutralize 5 c.c. of a solution of sulphuric acid and water, containing 40 milligrams H_2SO_4 per c.c. Now weigh or measure into any suitable closed vessel—a four-ounce sample bottle, for example—8.9 grams of the oil to be tested. To this add about two ounces No. 1, then add a few drops No. 2, and shake thoroughly. The color becomes yellow. Then add, from a burette graduated to c.c., solution No.3 until the color changes to red, and remains so after shaking. The acid is in proportion to the amount of solution (3) required. The best oils will require only from 4 to 30 c.c. to be neutralized and become red.

122. **Gumming or Drying.** —Gumming or drying is a conversion of the oil into a resin by a process of oxidation, and occurs after exposure of the oils to the air. In linseed and the drying oils it occurs very rapidly, and in the mineral oils very slowly.

Methods of Testing. —*Nasmyth's Apparatus.* —An iron plate six feet long, four inches wide, one end elevated one inch. Six or less different oils are started by means of brass tubes at the same instant from the upper end: the relative distances covered in a given time constitute a measure of the gumming property.

Bailey's Apparatus consists of an inclined plane, made of a glass plate, arranged so that it may be heated by boiling water. A scale and thermometer is attached to the plane. Its use is the same as the Nasmyth apparatus.

This effect may also be tested in an oil-testing machine by applying fresh oil, making a run, and noting the friction; then exposing the journal to the effect of the air for a time, and noting the increase of friction. In all cases a comparison must be made with some standard oil.

123. Density of Oils. — The density or specific gravity is usually obtained with a hydrometer (see Fig. 174) adapted for this special purpose, and termed an oleometer. The distance that it sinks in a vessel of oil of known temperature is measured by the graduation on the stem; from this the specific gravity of the oil may be found. The standard temperature for measuring of the density of oils is 60° F.

The density is often expressed in Beaumé's hydrometer-scale, which can be reduced to corresponding specific gravities as compared with water by Table 2 given in the Appendix.

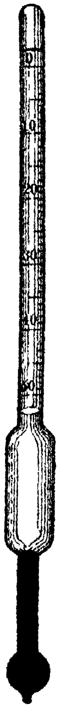

FIG. 174. HYDROMETER

Beaumé's hydrometer is graduated in degrees to accord with the density of a solution of common salt in water; thus, for liquids heavier than water the zero of the scale is obtained by immersing in pure water; the five-degree mark by immersing in a five-per-cent solution; the ten-degree mark in a ten-per-cent solution; etc. For liquids lighter than water the zero-mark is obtained by immersing in a ten-per-cent solution of brine; the ten-degree mark by immersing in pure water. After obtaining the length of a degree the stem is graduated by measurement.

The tendency to-day is to avoid the use of arbitrary scales, like the Beaumé, and to refer to the standard specific gravity scale.

The density may be found by obtaining the loss of weight of the same body in oil and in distilled water. The ratio of loss of weight will be the density compared with water.

It may also be obtained by weighing a given volume on a pair of chemical scales.

In connection with the methods of finding density above outlined it is well to remember that if a sample of oil is composite and has stood in a graduate for a considerable length of time the upper part of the oil column may be lighter than the lower, in which case the hydrometer may fail to give the correct result. This difficulty can usually be overcome by thorough stirring immediately before taking the reading.

The following table gives average results of density determinations for the most common oils.

SPECIFIC GRAVITY (AVERAGE) OF OILS AT 60° F.

Animal and Vegetable Oils.

Castor	0.966
Cotton-seed (refined)	0.923
Cod-liver (pure)	0.927
Lard (winter)	0.917
Linseed (raw)	0.929
Linseed (boiled)	0.941
Olive	0.914
Rape-seed (white winter)	0.914
Resin (third run)	0.988
Seal	0.924
Sperm (winter)	0.881
Tallow	0.904
Whale (winter)	0.925

Mineral Oils.

Rhigolene	0.622
Benzine	0.630 to 0.670
Gasolene	0.680 to 0.700
Illuminating oils (average)	0.780 to 0.860
Lubricating oils	0.860 to 0.960
Paraffine wax	0.890

124. — **Method of Finding Density.** — A. *With Hydrometer Thermometer and with Hydrometer Cylinder.*

1. Clean the cylinder thoroughly with benzine, then fill with distilled water. Set the whole in a water-jacket, and bring the temperature to 60° F. Obtain the reading of the hydrometer in the distilled water and determine its error.

2. Clean out the cylinder, dry it thoroughly, and fill with the oil to be tested; heat in a water-jacket to a temperature of 60° F., and obtain reading of hydrometer; also obtain reading, at temperatures of 40°, 80°, 100°, 125°, and 150°, and plot a curve showing relation of temperature and corrected hydrometer-reading.

Reduce hydrometer-readings, if in Beaumé degrees, to corresponding specific gravities, by table given in Appendix.

B. Weigh on a chemical balance the same volume of distilled water at 60° F., and of the oil at the same temperature; and compute the specific gravity.

C. Weigh the same metallic body by suspending from the bottom of a scale-pan of a balance: 1. In air; 2. In water; 3. In the oil at the required temperature. Carefully clean the body with benzine after immersing in the oil. The ratio of the loss of weight in oil to that in water will be the density.

125. Viscosity. — *Viscosity of oil* is closely related but not proportional to density. It is also closely related, and in many cases it is inversely proportional, to the lubricating properties. The relation of the viscosities at ordinary temperatures is not the same as for higher temperatures, and tests for viscosity should be made with the temperatures the same as those in use. The less the viscosity, consistent with the pressure to be used, the less the friction.

The viscosity test is considered of great value in determining the lubricating qualities of oils, and it is quite probable that by means of it alone we could determine the lubricating qualities to such an extent that a poor oil would not be accepted nor a good oil rejected. It is, however, in the present method of performing it, to be considered as giving comparative rather than absolute results.

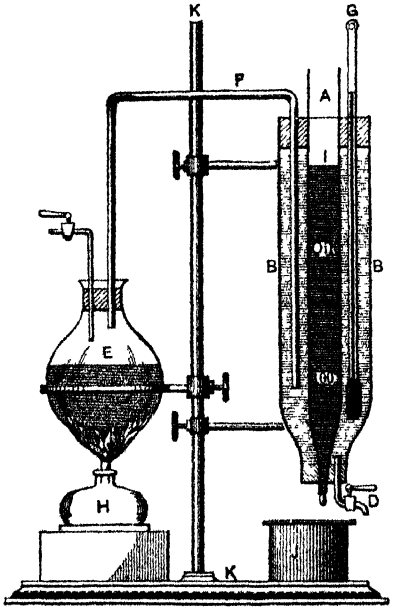

FIG. 175. — VISCOSIMETER.

There are several methods of determining the viscosity. It is usual to take the viscosity as inversely proportional to the rate of flow of the oil through a standard nozzle while maintaining a constant or constantly diminishing head and constant temperature, a comparison to be made with water or with some well-known oil, as sperm, lard, or rape-seed, under the same conditions of head and temperature.

Viscosimeter. — A pipette surrounded by a water-jacket, in which the water can be heated by an auxiliary lamp and maintained at any desired temperature, is often used as a viscosimeter. Fig. 175 shows the usual arrangement for this test. *E* is the heater for the jacket-water, *BB* the jacket, *A* the pipette, *G* a thermometer for determining the temperature of the jacket-water. The oil is usually allowed to run partially out from the pipette, in which case the head diminishes. Time for the whole run is noted with a stop-watch.

In the oil-tests made by the Pennsylvania Railroad Company the pipette is of special form, holding 100 c.c. between two marks,— one drawn on the stem, the other some distance from the end of the discharge-nozzle.

Tagliabue's Viscosimeter.— In Tagliabue's viscosimeter, shown in Figs. 176 and 177, the oil is contained in a basin *C*, and trickles downward through a metal coil, being discharged at the faucet on the side into a vessel holding 50 c.c. The oil is maintained at any desired temperature by heating the water in the vessel *B* surrounding the coil; cold water is supplied from the vessel A, as required to maintain a uniform temperature. The temperature of the oil is taken by the thermometer *D*.

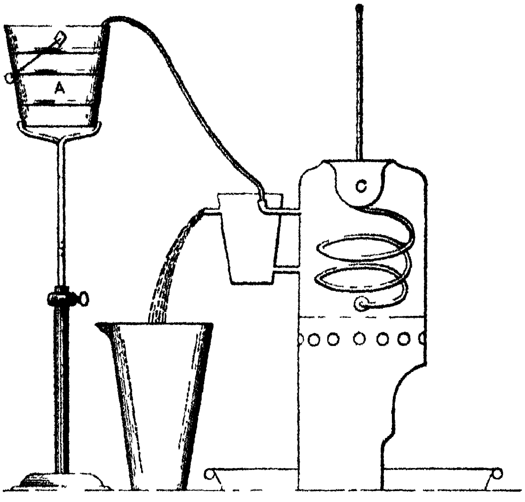

FIG. 176 — TAGLIABUE'S VISCOSIMETER.

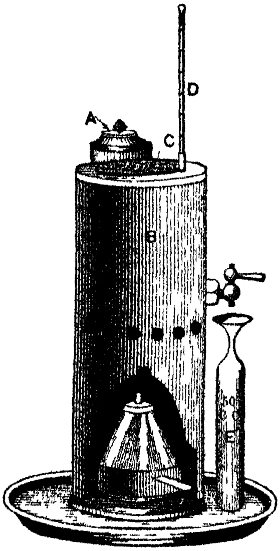

FIG. 177.— TAGLIABUE'S VISCOSIMETER.

Gibbs' Viscosimeter. — In the practical use of viscosimeters it is found that the time of flow of 100 c.c. of the same oil, even at the same temperature, is not always the same, — which is probably due to the change in friction of the oil adhering to the sides of the pipette.

To render the conditions which produce flow more constant, Mr. George Gibbs of Chicago surrounds the viscosimeter, which is of the pipette form, with a jacket of hot oil. A circulation of the jacket-oil is maintained by a force-pump. The oil to be tested is discharged under a constant head, which is insured by air-pressure applied by a pneumatic trough. The temperature of the discharged oil is measured near the point of discharge.

Perkins' Viscosimeter.—The Perkins' viscosimeter consists of a cylindrical vessel of glass, surrounded by a water or oil bath, and fitted with a piston and rod of glass. The edges of this piston are rounded, so as not to be caught by a slight angularity of motion. The diameter is one-thousandth of an inch less than that of the cylinder. In practice the cylinder is filled nearly full of the oil to be tested, and the piston inserted. The time required for the piston to sink a certain distance into the oil is taken as the measure of the viscosity.*

Stillman's Viscosimeter. —Prof. Thomas B. Stillman of Stevens Institute uses a conical vessel of copper, $6\frac{3}{8}$ inches in length and $1\frac{3}{4}$ inches greatest diameter, surrounded by a water-bath, and connected to a small branch tube of glass, which is graduated in cubic centimeters; the time taken for 25 c.c. to flow through a bottom orifice $\frac{3}{64}$ of an inch in diameter is taken as the measure of the viscosity, during which time the head changes from 6 to 5 inches. Prof. Stillman makes all comparisons with water, which is the most convenient and uniform standard. The temperature of the oil is taken at about the center of the viscosimeter.

Viscosimeter with Constant Head. —A form of viscosimeter which possesses the advantage of having a constant head for flow of oil regardless of the quantity in the instrument, as made by Tinius Olsen & Co. of Philadelphia, is shown in the next figure. It is simple in form and can be very readily cleaned. It is provided with a jacket, and oils may be tested at any temperature. This instrument is now the principal standard used in the Sibley College Laboratories.

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* See paper by Prof Denton, Vol IX., Transactions of Am Society of Mechanical Engineers.

Description.—*A*, Fig. 178, is a cup similar in construction to that of the kerosene reservoir of a students' lamp, with a capacity of about 125 c.c., and is surrounded with a jacket *D*, in which may be placed insulating materials or water to maintain a constant temperature while the oil is flowing; *C* is a thermometer-cup, to the bottom of which is secured a small cap containing the orifice *F*; *N* is a channel connecting the chamber containing *A* with *C*; *B* is one of four small tubes which admit air to the interior of the cup *A* and thus maintain atmospheric pressure on oil in it; this action secures a constant level of the surface of the oil in the cup *C* and the surrounding space, at the height of the lower opening in the tube *B*. *H* is a valve to retain oil in *A* while placing it into *D*. *M* is a bracket serving as a guide for valve-stem *K*.

The mechanism *L*, *G*, *G* is a device for opening and closing the orifice *F* readily, and is held in a closed position by spring catch *L*.

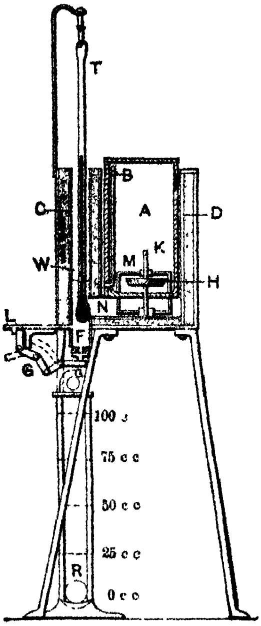

FIG. 178.
VISCOSIMETER.

The instrument is supported by three legs about eight inches in length.

Operation.—Withdraw cup *A*, fill it in an inverted position with the oil, hold valve *H* on its seat, reinvert *A* and place in position in the instrument. The latter operation raises valve *H* and the oil is allowed to flow out of *A* until chambers *N* and *C* are filled a little above lower opening of tube *B*. A beaker graduated in c.c.'s, of capacity of about 110 c.c., is placed under *F*; *L* is released and *G* allowed to drop, permitting oil to flow through *F* freely into the beaker. When oil in *C* falls below the bottom of tube *B*, air is admitted to the top of the oil in *A* and oil flows out until it again rises a little above the lower end of tube *B*, when flow out of *A* is stopped until the level again falls below *B*. This action continues throughout entire run, intermittently but so rapidly that a practically constant head is maintained at *F*.

In *C* a thermometer is suspended so that its bulb is immersed in the oil, by which means the temperature of oil can be observed immediately before flowing out of orifice *F*, which is essential in ascertaining the relative viscosity of the oil. The oil may be heated in the viscosimeter by applying a Bunsen burner, but it is usually more conveniently heated in a separate vessel until it has attained the proper temperature.

Other Forms of Viscosimeters. — A simple form of viscosimeter has been used with success by the author, consisting of a copper cup in form of a frustum of a cone, having dimensions as follows: bottom diameter 1.25 inches, top diameter 1.95 inches, depth 6 inches. The flow takes place through a sharp-edged orifice in the centre of the bottom $\frac{1}{16}$ inch in diameter. The whole height is 6½ inches. The instrument when made of copper requires a glass oil-gauge, showing the height of the oil in the viscosimeter. This should be connected to the viscosimeter 3 inches from the bottom. The time for the flow of 100 c.c. is taken as the measure of the viscosity, during which time the head changes from 6 to about 3.5 inches; the area of exposed surface diminishes at almost exactly the rate of decrease of velocity of flow, so that the fall of level is very nearly constant.

The comparative number of vibrations of a pendulum swinging freely in the air, and when immersed in an oil during a given time, is also said to afford a valuable means of determining the viscosity.

126. Method of Measuring Viscosity with Olsen Apparatus. — If the viscosimeter has a water-jacket fill it and arrange for the maintenance of the same at any desired temperature. This is most conveniently done by circulation from a water-bath. Fill the cup with the oil and insert in the instrument. Allow the oil to run out, noting accurately with the stop-watch the exact time required to discharge a given amount. Make determinations at 60°, 100°, and 150° F., two for each temperature. Clean the apparatus thoroughly at the beginning and end of the test, using benzine or alkali to remove any traces of oil.

The ratio of time of flow of a quantity of oil to time of flow of an equal quantity of water measures the relative viscosity of the given sample of oil to that of water at the given temperature.

127. Chill-point. — Cold tests are made to determine the behavior of oils and greases at low temperatures. By chill-point is meant the temperature of solidification. The method of test is to expose the sample while in a wide-mouthed bottle or test tube to the action of a freezing mixture, which surrounds the oil to be tested. Freezing mixtures may be made with ice and common salt, with ice alone, or with 15 parts of Glauber's salts, above which is a mixture of 5 parts muriatic acid and 5 parts of cold water. The temperature is read from a thermometer immersed in the oil. The chill point is to be found by first freezing and then determining the melting temperature.

Tagliabue's Cold-test Apparatus. — Tagliabue has a special apparatus for the cold test of oils shown in section in Fig. 179. The

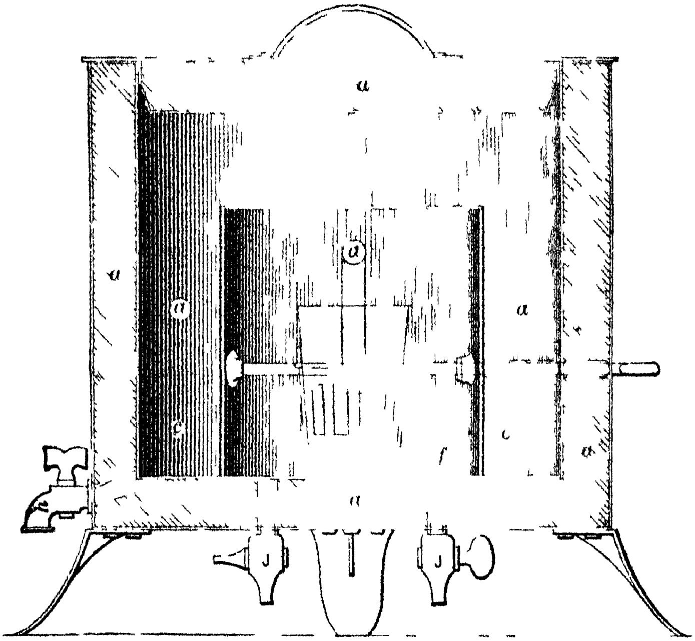

FIG. 179.—TAGLIABUE'S COLD-TEST APPARATUS.

oil is placed in the glass vessel, which is surrounded with a freezing mixture. The glass containing the oil can be rocked backward and forward, to insure more thorough freezing. A thermometer is inserted into the oil and another into the surrounding air-chamber, the oil is frozen, then permitted to melt, and the temperature taken.

In making this test considerable difficulty may be experienced in determining the melting-point since many of the oils do not suddenly freeze and thaw like water, but gradually soften, until they will finally run, and during this whole change the temperature will continue to rise. This is no doubt due to a mixture of various constituents with different melting-points. In such a case it is recommended that an arbitrary chill-point be assumed at the temperature that is indicated by a thermometer inserted in the oil, when it has attained sufficient fluidity to run slowly from an inverted test-tube. The temperature at the beginning and end of the process of melting is to be observed.

128. Method of Finding the Chill-point. — Pour the sample to be tested into a test-tube or other vessel, in which insert the thermometer; surround this with the freezing mixture, which may be composed of small particles of ice mixed with salt, with provision for draining off the water. Allow the sample to congeal, remove the test-tube or vessel from the freezing mixture, and while holding it in the hand stir gently with the thermometer. The temperature indicated when the oil is melted is the chill-point.

In case the operation of melting takes place over a range of temperature, note the temperature at the beginning and also at the end of the process of melting.

In report describe apparatus used and the methods of testing.

129. The Flash-point. — *The Flash-test* determines the temperature at which oils discharge by distillation vapors which may be ignited. The test is made in two ways.

Firstly. *With the open cup.* — In this case the oil to be tested is placed in an open cup of watch-glass form, which rests on a sand-bath. A thermometer is suspended with the bulb immersed in the oil. Heat is applied to the sand-bath, and as the oil becomes heated a lighted taper or match is passed at intervals of a few seconds over the surface of the oil, and at a distance of about one-half inch from it. At the instant of flashing the temperature of the oil is noted, which temperature is the "flash-point."

Fig. 180 shows Tagliabue's form of the open cup, in which heat is applied by a spirit-lamp to a water or sand bath surrounding the cup containing the oil.

The method of applying the match is found to have great influence on the determination of the flash-point, and should be distinctly stated in each case. When the vapor is heavier than air, a lower flash-point will be shown by holding near one edge of the cup.

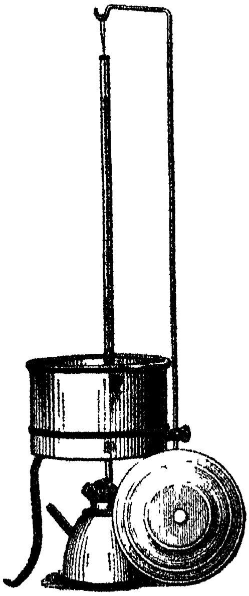

FIG. 180. — OPEN CUP.

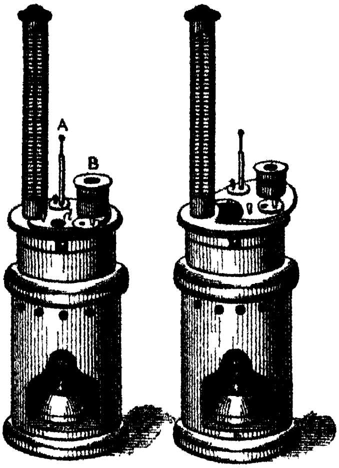

FIG. 181. — CLOSED CUP FOR FLASH-POINT.

Secondly. *With the closed oil-cup.* — Fig. 181 is a view of Tagliabue's closed cup for obtaining the flash-point; in this instrument the oil is heated in a sand-bath above a lamp. The thermometer gives the temperature of the oil. The match is applied from time to time at the orifice *d*, which in the intervals can be covered by a slide.

The open cup is generally preferred to the closed one as giving more uniform determinations, and it is also more convenient and less likely to explode than the closed one.

130. Method of Testing for Flash-point. — Put some dry sand or water in the outer cup and some of the oil to be tested in the small cup. Light the lamp and heat the oil gently — at the rate of about 50° F. in a quarter of an hour. At intervals of half a minute after a temperature of 100° F. is attained, pass a lighted match or taper slowly over the oil at a distance of one-half inch from the surface. The reading of the thermometer taken immediately before the vapor flashes is the temperature of the flash-point.

With the closed cup the method is essentially the same. The lighted taper is applied to the tube leading from the oil vessel, the valve being opened only long enough for this purpose.

131. **Burning-point.** — The *burning-point* is determined by heating the oil to such a temperature that when the match is applied as for the flash-test the vapors above the surface burn continuously. The reading of the thermometer just before the match is applied is the burning-point.

132. **Method of Testing for Burning-point.** — The burning-point is found in the same manner as the flash-point, with the open cup, the test being continued until the oil takes fire when the match is applied. The last reading of the thermometer before combustion commences is the burning-point.

133. **Volatility.** — Mineral oil will lose weight by evaporation, which may be ascertained by placing a given weight in a watch-glass and exposing to the heat of a water-bath for a given time, as twelve hours. The loss denotes the existence of volatile vapors, and should not exceed 5 per cent in good oil. Other oils often gain weight by absorption of oxygen.

134. **Coefficient of Friction. — Oil-testing Machines.** — *Measurements of the coefficients of friction* are made on oil-testing machines, of which various forms have been built. These machines are all species of dynamometers, which provide (1) means of measuring the total work received and that delivered, the difference being the work of friction; or (2) means of measuring the work of friction directly. Machines of the latter class are the ones commonly employed for this especial purpose.

Rankine's Oil-testing Machine. — Rankine describes two forms of apparatus for testing the lubricating properties of oil and grease.

I. *Statical Apparatus.* — This consists of a short cylindrical axle, supported on two bearings and driven by pulleys at each end. In the middle of the axle a plumber-block was rigidly connected to a mass of heavy material, forming a pendulum. The lubricant to be tested was inserted in the plumber-block attached to the pendulum, and the coefficient of friction determined by the deviation of the pendulum from a vertical. In this machine the axle was provided with reversing-gears, so that it could be driven first in one direction and then in the opposite. With this class of machine, if r equal the radius of the journal, R the effective arm of the pendulum, P the total force acting on the journal, ϕ the angle with the vertical, we shall have the product of the force P into the arm $R \sin \phi$ equal to the moment of resistance Fr. That is,

$$Fr = PR \sin \phi,$$

from which

$$f = \frac{F}{P} = \frac{PR \sin \phi}{Pr} = \frac{R \sin \phi}{r}.$$

II. *Dynamic or Kinetic Apparatus.* — In this case a loose fly-wheel of the required weight is used instead of the pendulum. The bearings of journals and of fly-wheel are lubricated; then the machine is set in motion at a speed greater than the normal. The driving-power is then disengaged, and the fly-disk rotates on the stationary axis until it comes to rest. The coefficient of friction is obtained by measuring the retardation in a given time.

Let the initial number of turns per second $= n_1$,
the final number of turns per second $= n_2$,
ω_1 and ω_2 = the corresponding angular velocities,
W = weight of the wheel,
k = radius of gyration,
$I = \frac{W}{g} k^2$ = moment of inertia of the wheel,
r = radius of the journal, and
t = time of retardation in seconds.

Then the loss in kinetic energy

$$= \frac{1}{2} \frac{W}{g} k^2 (\omega_1^2 - \omega_2^2).$$

The work of friction during t seconds

$$= F \cdot 2 \pi r \left(\frac{n_1 + n_2}{2}\right) t.$$

These two expressions must be equal, and, remembering that $\omega = 2 \pi n$, the equation will reduce to

$$F = \frac{2 \pi (n_1 - n_2)}{rt} \frac{Wk^2}{g}.$$

Now $F = fW$, where f is the coefficient of friction.

Hence finally
$$f = \frac{2\pi(n_1 - n_2)}{rt}\frac{k^2}{g}.$$

Thurston's Standard Oil-testing Machine.—This machine permits variation in speed and in pressure on the journal; it also affords means of supplying oil at any time, of reading the pressure on the journal, and the friction on graduated scales attached to the machine.

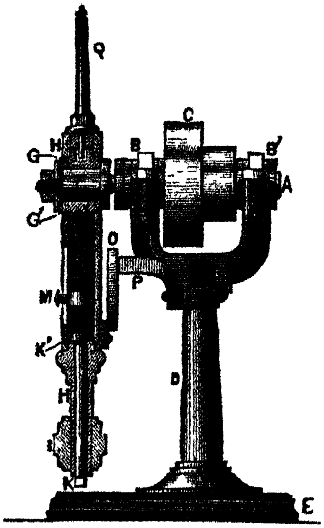

Fig. 182.—Section of Thurston's Oil-testing Machine.

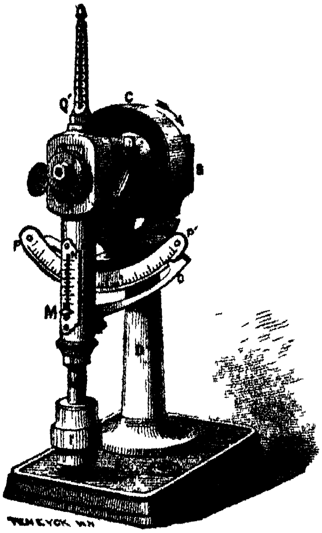

Fig. 183.—Perspective View of Thurston's Oil-testing Machine.

This machine, as shown in the following cuts, Figs. 182 and 183, consists of a cone of pulleys *C*, for various speeds, fastened on shaft *A* between two bearings *B*, *B'*. The shaft carries an overhanging journal, *F*, about which the pendulum *H* swings. The latter is supported by brasses which are adjustable and which may be set to exert any given pressure by means of an adjusting screw *K'*, acting on a coiled spring within the pendulum. The pressure so exerted can be read directly on the scale *M*, attached to the pendulum; a thermometer *Q* in the upper brass gives the temperature of the bearings. The deviation of the pendulum is measured by a graduated arc *PP'*, fastened to the frame of the machine. The graduations of the pendulum scale *M* show on one side the total pressure on the journal *P*, and on the other the pressure per square inch p; those on the fixed scale PP' show the total friction F; this, divided by the total pressure P, gives f, the coefficient of friction.

From the construction of the machine, it is at once perceived that the pressure on the journal is made up of equal pressures due to action of the spring on upper and lower brasses, and of the pressure due to the weight of the pendulum, which acts only on the upper brass. This latter weight is often very small, in which case it can be neglected without sensible error.

Thurston's Railroad Lubricant-tester.—The Thurston machine is made in two sizes; the larger one, having axles and bearings of the same dimensions as those used in standard-car construction, is termed the "Railroad Lubricant-testing Machine." A form of this machine is shown in the following cut, arranged for testing with a limited supply of lubricant. (See Fig. 184.)

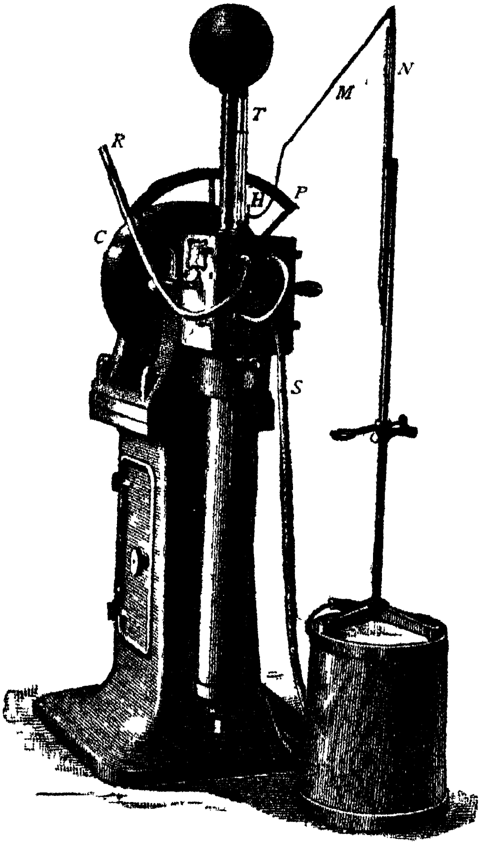

FIG 184 —THURSTON'S RAILROAD LUBRICANT TESTING MACHINE.

Explanation of symbols:

T, thermometer, giving temperature of bearings.

R, S, rubber tubes for circulation of water through the bearings.

N, burette, furnishing supply of oil.

M, siphon, controlling supply of oil.

P, candle-wicking, for feeding the oil.

H, copper rod, for receiving oil from P.

The railroad testing-machine, which is shown in section in Fig. 185, differs from the Standard Oil-testing Machine principally in the construction of the pendulum. This is made by screwing a wrought-iron pipe *J*, which is shown by solid black shading in Fig. 185, into the head *K*, which embraces the journal and holds the bearings ***a***,***a′*** in their place. In this pipe a loose piece *b* is fitted, which bears against the under journal-bearing *a′*. Into the lower end of the pipe *J* a piece *cc* is screwed, which has a hole drilled through the center, through which a rod *f* passes, the upper end of which is screwed into a cap *d*; between this cap and the piece *cc* a spiral spring is placed. The upper end of the rod bears against the piece *b*, which in turn bears against the bearing *a′*. The piece *b* has a key *l*, which passes through it and the pipe *J*. This key bears against a nut *o*, screwed on the pipe. By turning the nut *o* the stress on the

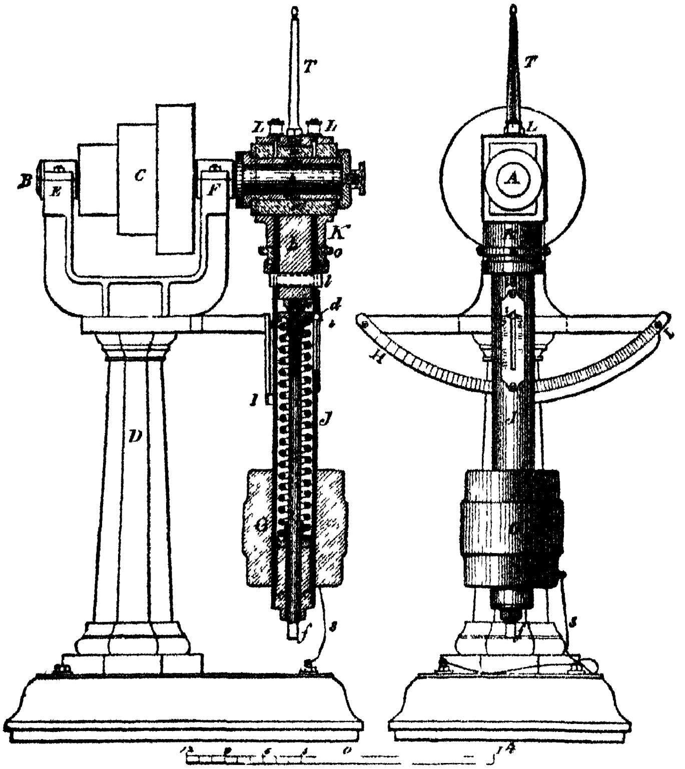

Fig. 185.—Section of Railroad Lubricant-testing Machine.

journal produced by screwing the rod *f* can be thrown on the key *l*, and the bearing relieved of pressure, without changing the tension on the spring. A counterbalance above the pendulum is used when accurate readings are desired. The " brasses " are cast hollow, and when necessary a stream of water can be passed through to take up the heat, and maintain them at an even temperature.

The graduations on the machine show on the fixed scale, as in the standard machine, the total friction; and on the pendulum, the total pressures (1) on the upper brasses, (2) on the lower brasses, and (3) the sum of these pressures.

Theory of the Thurston Oil-testing Machines.

The mathematical formulæ applying to these machines are as follows: Let P equal the total pressure on the journal; p the pressure per square inch on projected area of journal; T the tension of the spring; W the weight of the pendulum; r the radius of the journal; R the effective arm of the pendulum; θ the angle of deviation of the pendulum from a vertical line; F the total force of friction; f the coefficient of friction; l the length of bearing-surface of each brass.

Since in this machine both brasses are loaded, the projected area of the journal bearing-surface is $2\ (2\ r)l = 4\ lr$. We shall evidently have

$$P = 2\ T + W, \tag{1}$$

$$p = \frac{P}{4\,lr} = \frac{2\ T + W}{4\,lr}. \tag{2}$$

By definition $f = F \div P$.

Since the moment of friction is equal to the external moment of forces acting,

$$Fr = Pfr = f\,(2\ T + W)\,r = WR \sin\theta. \tag{3}$$

From which

$$f = \frac{F}{P} = \frac{WR \sin\theta}{rP}. \tag{4}$$

In the machines $WR \sin\theta \div r$ is shown on the fixed scale, and the graduations will evidently vary with $\sin\theta$, since $WR \div r$ is constant.

P, the total pressure, is shown on the scale attached to the pendulum.

In the standard machine the weight of the pendulum is neglected, and $P = 2T$; but in the railroad oil-testing machine the weight must be considered, and $P = 2T + W$, as in equation (1).

Constants of the Machine.

As the constants of the machine are likely to change with use, they should be determined before every important test, and the final results corrected accordingly.

1. To determine the constant WR, swing the pendulum to a horizontal position, as determined by a spirit-level; support it in this position by a pointed strut resting on a pair of scales. From the weight, corrected for weight of strut, get the value of WR; this should be repeated several times, and the average of these products obtained.

2. Obtain the weight of the pendulum by a number of careful weighings.

3. Measure the length and radius of the journal; compute the projected bearing-surface $2\,(2\,lr)$.

4. Compute the constant $\frac{WR}{r}$, which should equal twice the reading of the arc showing the coefficient of friction when the pendulum is at an angle of 30°, since sine of 30° equal $\frac{1}{2}$.

Riehlé's Oil-testing Machine. — This machine consists of an axle revolving in two brass boxes, which may be clamped more or less tightly together. The machine as shown in Fig. 186 has two scale-beams, — the lower one for the purpose of weighing the pressure put upon the journal by the hand-screw on the opposite side of the machine, the upper one for measuring the tendency of the journal to rotate. The upper scale-beam shows the total friction, or coefficient of friction, as the graduations may be arranged. A thermometer gives the temperature of the journal; a counter the number of revolutions.

The Olsen-Cornell Oil-testing Machine. — In both the Thurston and Riehlé machines pressure is applied to both halves of the bearing and there are consequently two points of maximum pressure on the journal. This is an unusual condition in actual practice and in this respect these machines fail to represent true conditions, although the results obtained with them can of course be compared among themselves. In the Cornell testing machine, Fig. 187, the test bearing is a block resting on the top of the test journal, which in turn is supported by a bearing on each side. The driving pulley is placed next to the test journal. The test bearing is held by a yoke which is supported on knife-edges in the axis of the test journal. Pressure is applied to the bearing by pulling down the yoke by means of a strong helical spring contained in the cylindrical case near the base of the machine. The force of friction tends to swing the yoke about its knife-edges, which tendency is balanced by means of the scale beam. The total pressure is determined by calibrating the load spring, the figures being given on a scale attached to the spring case.

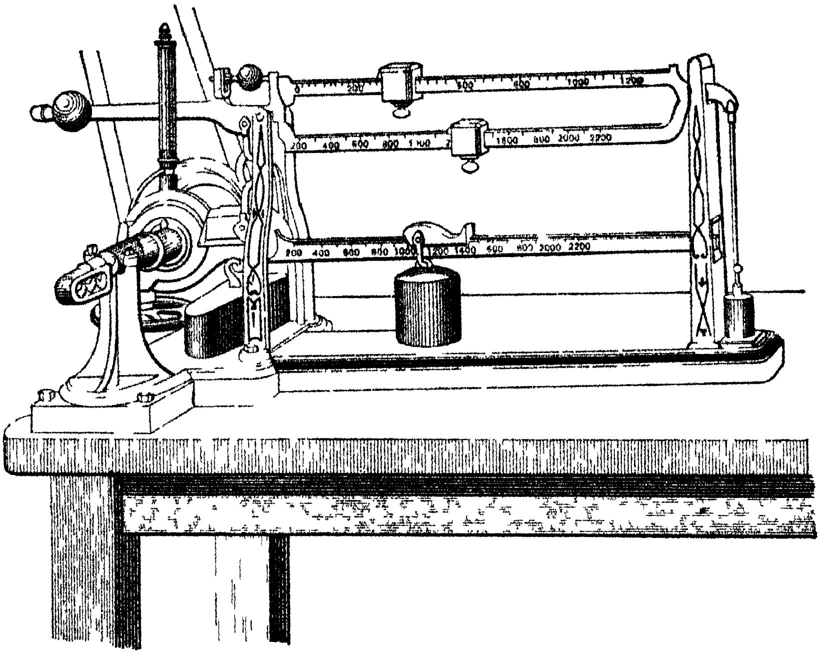

Fig. 186.—Riehlé's Oil-testing Machine.

As in the other machines, the coefficient of friction is equal to the force of friction, in this case read directly from the beam, divided by the total pressure. A test bearing of the kind used in this machine represents more nearly the conditions existing in an actual bearing.

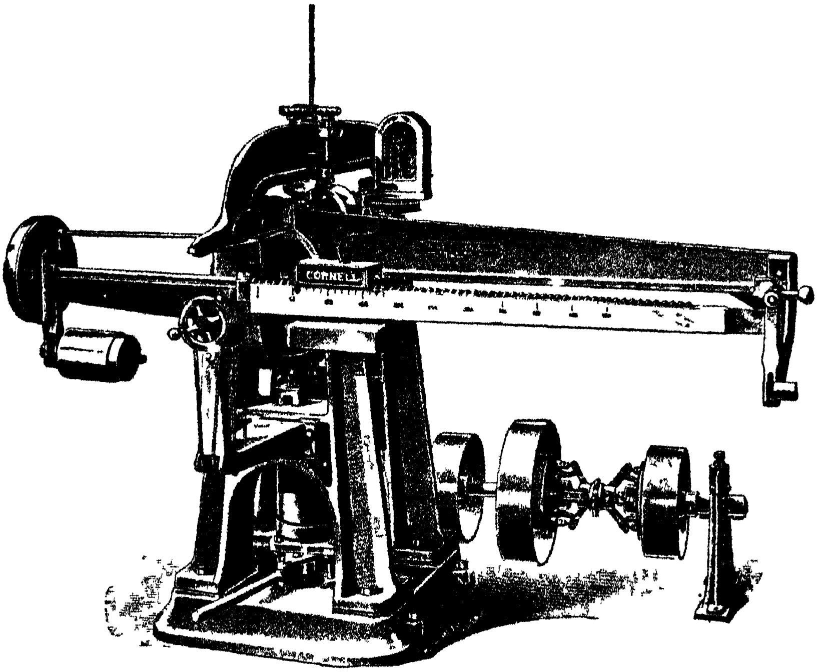

Fig. 187.—Olsen-Cornell Oil-testing Machine.

135. Directions for Obtaining Coefficient of Friction with Thurston's Oil-testing Machines. —*Cleaning.* —In the testing of oils great care must be taken to prevent the mixing of different samples, and in changing from one oil to another the machine must be thoroughly cleaned by the use of alkali or benzine.

In the test for coefficient of friction the loads, velocity, and temperature are kept constant for each run; the oil-supply is sufficient to keep temperature constant, the journals being generally flooded. The load is changed for each run.

The following are the special directions for the test of *Coefficient of Friction*, as followed in the Sibley College Engineering Laboratory.

Apparatus. —Thurston's Standard Lubricant-testing Machine; thermometer; attached revolution-counter.

Method. — Remove and thoroughly clean the brasses and the steel sleeve or journal by the use of gasolene. Determine the constants of the machine as explained in Article 134, measure the projected area of journal bearing-surface, and the weight and moment of the pendulum. Ascertain the error, if any, in the graduation of the machine, and correct the results obtained accordingly. Put the sleeve on the mandrel; place the brasses in the head of the pendulum and see that the pressure spring is set for zero and pressure as indicated by the pointer on the scale. Slide the pendulum carefully over the sleeve, put on the washer, and secure it with the nut. See that the feeding apparatus is in running order. Belt up the machine for the high speed and throw on the power, at the same time supplying the oil at a rate calculated to maintain a free supply. By deflecting the pendulum and using a wrench on the nut at the bottom increase the pressure on the brasses gradually until the pointer indicates 50 pounds per square inch.

Make a run at this pressure, and also for pressures of 100, 150, and 200 pounds; but do not in general permit the maximum pressure in pounds per square inch to exceed $44{,}800 \div (v + 20)$. Begin by noting the time and the reading of the revolution-counter; take readings, at intervals of one minute, of the arc and the temperature until both are constant. At the end of the run read the revolution-counter and note the time.

The velocity, v, of the rubbing surface in feet per minute should be computed from the number of revolutions and circumference of the journal.

Make a second series of runs, with constant pressure and variable speed.

In report of the test state clearly the objects, describe apparatus used and method of testing.

Tabulate data, and make record of tests on the forms given.

Draw a series of curves on the same sheet, showing results of the various tests as follows:

1. With total friction as abscissæ, and pressure per square inch as ordinates; for constant speed.

2. With coefficient of friction as abscissæ, and pressure per square inch as ordinates; for constant speed.

3. With coefficient of friction as abscissæ, and velocity of rubbing in feet per minute as ordinates; pressure constant.

136. Instructions for Use of Thurston's R. R. Lubricant-tester. —Follow same directions for coefficient of friction-test as given for the standard machine, applying the pressure as explained in Article 135.

Water or oil of any desired temperature can be forced through the hollow boxes by connecting as shown in Fig. 184, and the temperature of the bearings thus maintained at any desired point. With this arrangement the machine may be used for testing cylinder-stock, as explained in directions for using Boult's machine. The concise directions are:

1. Clean the machine.
2. Obtain the constants of the machine; do not trust to the graduations.
3. Make run under required conditions, which may be with each rate of speed.
    - *a.* With flooded bearings, temperature variable.
    - *b.* With flooded bearings, temperature regulated by forcing oil or water through hollow brasses.
    - *c.* Feed limited, temperature variable or temperature regulated.

In all cases the object will be to ascertain the coefficient of friction.

137. Durability of Lubricants. —In this case the amount of oil supplied is limited, and it is to be used for as long a time as it will continue to cover and lubricate the journal and prevent abrasion. To give satisfactory results, this requires a limited supply or a perfectly constant rate of feed, an even distribution of the oil, and the restoration of any oil that is not used to destruction; these requirements present serious difficulties, and present methods do not give uniform results.* The method at present used is to consider the endurance or durability proportional to the time in which a limited amount, as one-fourth c.c., will continue to cover and lubricate the journal without assuming a pasty or gummy condition, and without giving a high coefficient of friction. The average of a number of runs is taken as the correct determination. In this test care must be taken not to injure the journal.

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* See paper by Professor Denton, Vol. XI., p. 1013, Transactions of American Society of Mechanical Engineers.

The time or number of revolutions required to raise the temperature to a fixed point—for instance, 160° F.—is in some instances considered proportional to the durability.

138. Durability-testing Machines.—The Ashcroft and the Boult machines are especially designed for determining the durability of oils—from the former by noting the rise in temperature, from the latter by noting the change in the coefficient of friction. The difficulty of properly making this test no doubt lies in the loss of a very slight amount of oil from the journals, which is sufficient, however, to make the results very uncertain.

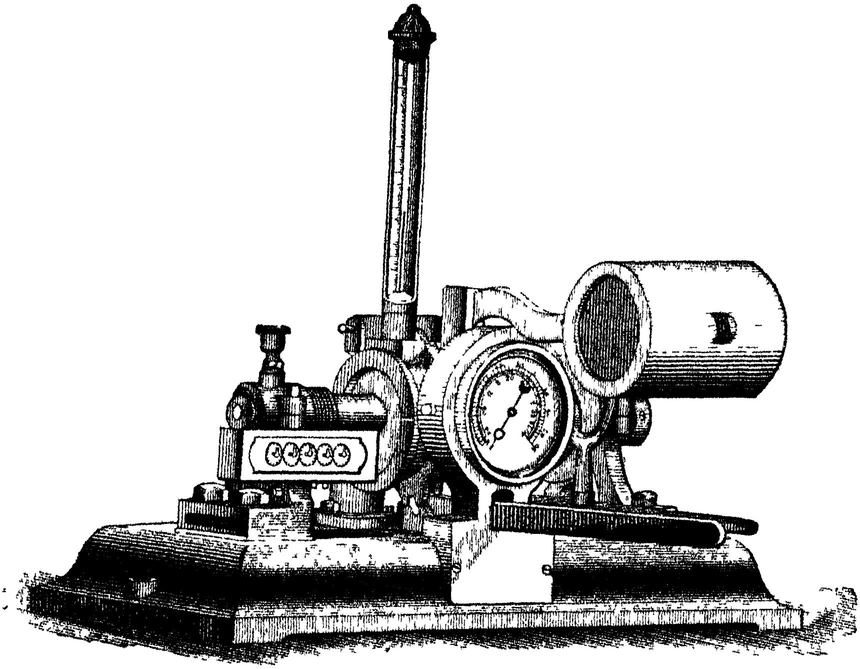

FIG. 188.—ASHCROFT OIL-TESTING MACHINE.

Ashcroft's Oil-testing Machine.—This machine (Fig. 188) consists of an axle revolving in two brass boxes; the pressure on the axle is regulated by the heavy overhanging counterpoise shown in the engraving. The tendency to rotate is resisted by a lever which is connected to the attached gauge. The gauge is graduated to show coefficient of friction.

The temperature is taken by an attached thermometer, and the number of revolutions by a counter, as shown in the figure.

In this machine the weights and levers are constant, the variables being the temperature and coefficient of friction.

It is used exclusively with a limited supply of oil, the value of the oil being supposed to vary with the total number of revolutions required to raise the temperature to a given degree — for instance, to 160° F.

Boult's Lubricant-testing Machine. — This machine, designed by W. S. Boult of Liverpool, is a modification of the Thurston oil-tester, yet it differs in several essential features. A general view of the machine is shown in Fig. 189, and a section of its boxes and the surrounding bush in Fig. 190. The machine is designed to accom-

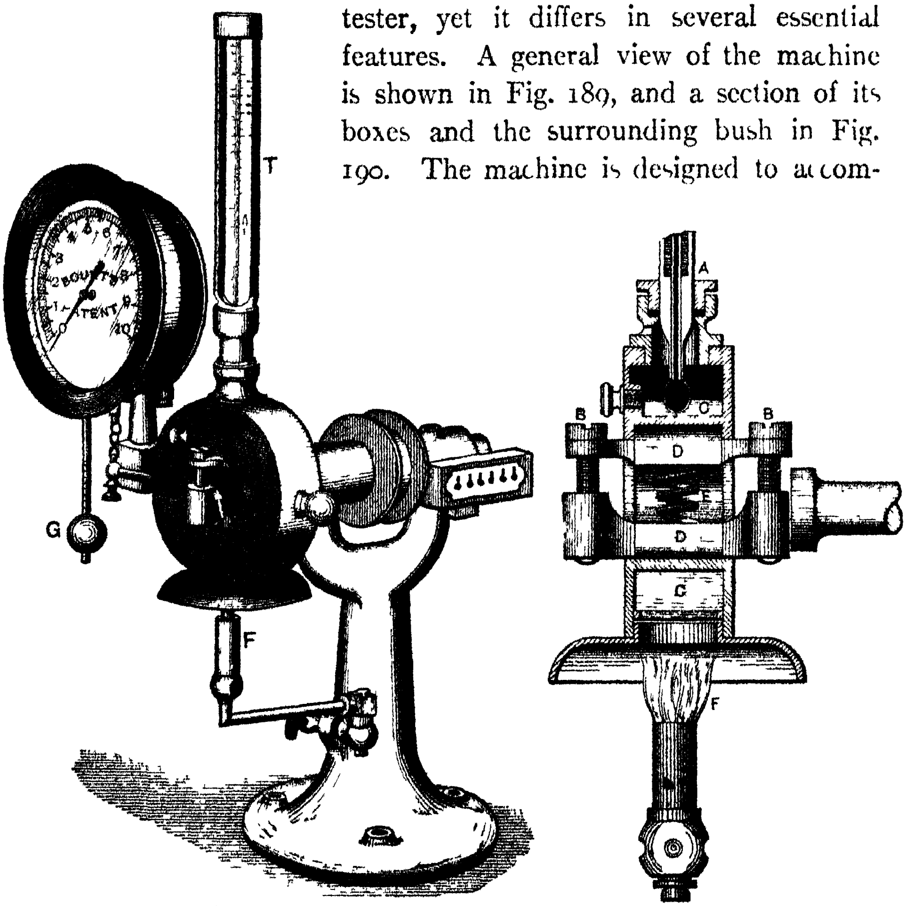

Fig 189. Fig 190

Boult's Lubricant-testers.

plish the following purposes: 1. Maintaining the testing journal at any desired temperature. 2. Complete retention on the rubbing surfaces of the oil under test. 3. Application of suitable pressure to the rubbing surfaces. 4. Measurement of the friction between the rubbing surfaces.

To secure the complete retention of the oil, a complete bush with internal flanges is used instead of the brasses employed in other oil-testing machines. On the inside of the bush is an expanding journal *DD*, Fig. 190, the parts of which are pressed outward against the surrounding bush by the spring *E*, or they may be drawn together by the set-screws *BB*, compressing the spring *E*. A limited amount of oil is fed from a pipette or graduated cylinder onto the journal, with the bush removed. This oil, it is claimed, will be maintained on the outer surface of the journal and on the interior surface of the metallic bush, so that it may be used to destruction. The bush is hollow, and can be filled with water, oil, or melting ice and brine.

The oil to be tested can be maintained at any desired temperature by a burner *F*, which heats the liquid *CC* in the surrounding bush. The temperature of the journal can be read by a thermometer whose bulb is inserted in the liquid *CC*.

The friction tends to rotate the bush; this tendency is resisted by a lever connected by a chain to an axis carrying a weighted pendulum *G*, Fig. 189.

The motion of the pendulum is communicated by gearing to a hand, passing over a dial graduated to show the total friction on the rubbing surfaces.

The formulæ for use of the instrument would be as follows: Let f equal coefficient of friction; G the weight of the bob on the pendulum, R its lever-arm; α the angle made by the pendulum with the vertical; a the length of the connecting lever; c the radius of the axis to which the pendulum is attached; r the radius of the journal; A the projected area of the journal; P the total pressure on the journal. Then

$$\frac{a}{r} \cdot \frac{R}{c} \cdot G \sin \alpha = fAP,$$

from which

$$f = \frac{aGR \sin \alpha}{rcAP} = \frac{\sin \alpha}{P} \times \text{a constant.}$$

In this instrument the total pressure P is usually constant and equal to 68 pounds, so that the graduations on the dial must be proportional to sin α.

If the graduations are correct, the coefficient is found by dividing the readings of the dial by P (68 pounds). The work of friction is the product of the total space travelled into the total friction, and this space in the Boult instrument is two-thirds of a foot for each revolution, or two-thirds of the number of revolutions.

The instrument cannot be used with a constant feed of oil, nor can the pressures be varied except by changing the spring E.

139. Directions for Durability-test of Oils with Boult's Oil-testing Machine. — To fill cylindrical oil-bath, take out the small thumb-screw and insert a bent funnel. Pour in oil — any sort of heavy oil may be used — until it overflows from the hole in which funnel is inserted, and replace thumb-screw.

1. See that the friction surfaces are perfectly clean. These can be examined by tightening the set-screws in order to compress the spring. This will enable the cylindrical bath to be lifted away. After seeing that the surfaces are perfectly clean, pour on a measured quantity of the lubricant to be tested, and set the bath in position. Slacken set-screws so as to allow the spring to have full pressure. The set-screws should not be removed entirely when slackening.

2. Light the Bunsen burner.

3. Heat the oil-jacket until the temperature at which the oil is to be used is reached.

4. Read revolution-counter, start machine and note initial friction reading. Continue run until friction has increased 50 per cent above its initial value. In some cases it is preferable to run the tester until there is a rise of 100 per cent of the friction first indicated. There does not appear to be any advantage in going beyond this, as the oil is then practically unfit for further use, and there is danger of roughening the friction surfaces.

5. When it is considered desirable to ascertain the distance travelled by the friction surfaces during a test, read off the counting-indicator before and after the test, subtract the lesser from the greater total, and the difference will represent the number of revolutions made during the test. As the friction surfaces travel two-

thirds of a foot during each revolution, the number of feet travelled is arrived at by simply deducting from the number of revolutions made, one-third thereof.

The value of the oil is proportional to the number of feet travelled by the rubbing surfaces.

The speed at which the tester should be run should be about five to six hundred revolutions per minute. For high-speed engine-oil the speed may be increased to about a thousand per minute.

140. Friction of Ball- and Roller-bearings. — Within recent years ball- and roller-bearings have come into great prominence because of their low coefficient of friction and their capacity to endure abuse of various kinds. The common tests of such bearings may be divided into two classes:

1. Tests made to determine useful life of bearing under given load, and

2. Tests made to determine the coefficient of friction under different conditions.

For the purpose of the first test any machine by means of which a known load can be maintained on the bearing to be tested and which will record the total number of revolutions up to the point of failure, may be made to give satisfactory results.

A machine for the second test is, however, much more difficult of construction, as the friction loss to be measured is relatively very small, necessitating light, accurate parts, while the loads applied may be great, calling for stiff and heavy mechanism. Machines have been built along two distinct lines and the success or failure of either type depends to a great extent upon conditions.

In the first type several similar bearings of the type to be tested support a rotating member which is driven by an accurately calibrated motor. Means are provided for applying loads as desired and the test-bearing friction loss is found from the electrical input corrected for motor losses. The total friction loss divided by the number of bearings is assumed to be the loss due to each bearing.

The second type of machine operates in much the same way as the Thurston and Riehlé machines just described, the rotating part of bearing and shaft being driven in any convenient way and the frictional resistance being determined by measuring or weighing the tendency of the outer or stationary parts of the bearing to turn.

A very complete machine of this type was recently developed by Henry Hess of Philadelphia, and described in a paper before the A. S. M. E., in 1908. This machine is capable of measuring the bearing capacity and the frictional resistance of a ball- or roller-bearing loaded radially or axially or both. The line drawings, Fig. 191 and Fig. 192, show the method of operation.

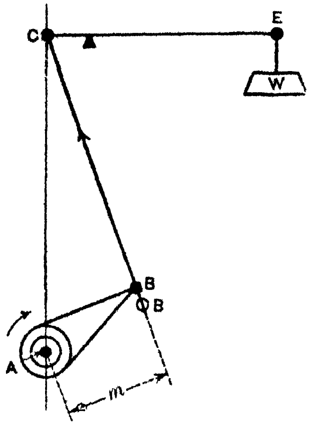

FIG. 191.

In Fig. 191, *A* is the bearing to be tested, mounted on the end of a shaft. The outer member is held in a strap *AB*, hinged to the rod *BC*, which is, in turn, hinged to a weighing system represented by *CE*. With the bearing at rest the points *A*, *B* and *C* are in a vertical line, but if the shaft is rotated in the direction of the arrow the various links assume a position similar, although much exaggerated, to that shown in the figure. The deflection of the point *B*, or any similar point *B′*, is a measure of the tendency to turn, or of the journal friction, and the machine is so arranged that this deflection may be accurately read to 0.0001 inch by means of a microscope and cross-hairs.

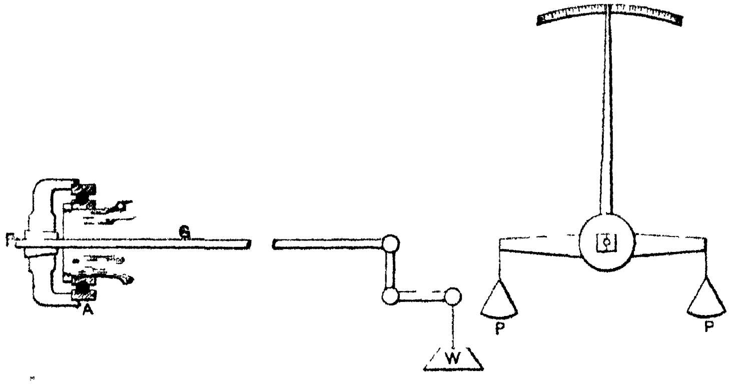

FIG. 192.

Fig. 192 shows the arrangement for applying axial loads. *G* is a wire which serves to draw the yoke *F* against the bearing with any desired force and also resists the tendency of the yoke to turn by twisting to such a point that its torsional resistance is sufficient for the purpose. The amount of frictional resistance can be measured by dropping weights into one or other of the pans *P* so as to bring the pointer which is carried by the yoke back to the zero reading.

When the journal is submitted to combined radial and thrust loads the deflection of the point *B*, Fig. 191, measures the total friction.

141. Forms for Report. — The following are the forms used in Sibley College for data and results of lubricant-tests:

SIBLEY COLLEGE, DEPARTMENT OF EXPERIMENTAL ENGINEERING.

VISCOSITY-TEST

Kind of Oil.. . . . . Date.. 191
Received from . . . .
Color. .. . .
Specific Gravity at... .. . .° F., Beaumé Scale . . . .° B.
" " " . . . . " Water as 1.000 .. . . . .
Ash. . . . . . % Tar . . . . . .% Chill-point .° F.
Flashing-Pt.. .. . .° F. Loss at ° F for 3 hours.. .. %
Burning-Pt.... . . .° F. Acid.. . . . .
Observers.. . . .

RESULTS OF VISCOSITY TEST.

No	Time of Flow of ..... C.C. in Sec			Temperature Deg. F.	Viscosity.	
	Sample.	Lard Oil.	Water		Water 1.00.	Lard Oil 1.00
1						
2						
3						
4						
5						
6						
7						
8						
9						
10						

OBSERVATIONS OF VISCOSITY-TEST.

No.	1	2	3	4	5	6	7	8	9	10
Temperature at beginning of run.										
Temperature at end of run.										
Average temperature.										
Time of flow of c.c. in seconds.										

DEPARTMENT OF EXPERIMENTAL ENGINEERING, SIBLEY COLLEGE, CORNELL UNIVERSITY.

FRICTION OF BEARING METALS AND LUBRICANTS.

Name of Lubricant.....

Bearing Metal.......

Journal.......... Diam.........

Bearing-surface: Length ...... Width.............. Area..........

Moment of Pendulum.......

Observers...... Date........

No. of test				
Pressure on journal, lbs. total				
Pressure on journal. lbs. per sq. in.....				
Method of lubrication				
Minimum coefficient of friction.....				
Maximum temperature of journal				
Temperature of room				
Elevation of temperature of journal above room				
Revolutions of journal per min.				
Feet travelled by rubbing-surface per min. ..				

Time.	Revolutions per min.	Temperature of Journal	Total Friction.	Coefficient of Friction.	Time.	Revolutions per min.	Temperature of Journal.	Total Friction.	Coefficient of Friction.

CHAPTER X.

MEASUREMENT AND TRANSMISSION OF POWER.

142. Definitions. — *Power is the rate of doing work,* while work is measured by the product of force times distance passed through. If we let P equal a force, l the distance passed through by the point of application of this force, and t the time in seconds required to pass through the distance l, then

$$\text{work} = P \times l \tag{1}$$

and

$$\text{power} = P \times \frac{l}{t}. \tag{2}$$

The factor $\frac{l}{t}$ is the distance passed through per second and is called the velocity.

If work is done by a rotating torque, Eqs. (1) and (2) are easily modified. Thus suppose that the force P acts at an arm a from the center of rotation and that the number of revolutions made in t seconds is n. Then the distance passed through by the point of application of the force P during n revolution is

$$l = 2\,\pi a n,$$

and as before,

$$\text{work} = P \times 2\,\pi a n, \tag{3}$$

while

$$\text{power} = P \times \frac{2\,\pi a n}{t}. \tag{4}$$

Eq. (4) may also be written

$$\text{power} = Pa \times \frac{2\,\pi n}{t} = Pa\omega, \tag{5}$$

in which ω = angular velocity (radians) in unit time.

In English units, P is expressed in pounds and l and a in feet, so that the *unit of work* is the foot-pound and the *unit of power* the foot-pound per unit time. The latter unit is too small for ordinary commercial use and hence the common power unit is the arbitrarily chosen *horse power*, which is equivalent to 550 ft.-lbs. per second or 33,000 ft.-lbs. per minute. The commercial *unit of electrical power* is the *kilowatt*, which is equivalent to 1.34 H.P., 1 H.P. being equal to .746 K.W. The horse power unit of the *metric* system is equal to 75 kilogram-meters per second. The metric horse power is therefore only 98.63 per cent of the English horse power.

143. Measurement of Power. — It must be evident from a study of Eqs. (2) and (5) that the measurement of power resolves itself into the measurement of force and linear velocity, or torque and angular velocity. The measurement of the velocity factor is usually very simple; that of the force or torque factor is, however, often quite complicated. Instruments and machines used to measure power are collectively known as *dynamometers*. There are two distinct types, the *absorption* and the *transmission* dynamometer. In the former, as the name indicates, the power is absorbed, or in a sense destroyed, being generally converted into heat which is dissipated, although the electric generator may be considered a form of absorption dynamometer. Transmission dynamometers, on the other hand, except for the losses due to friction in the dynamometer itself, transmit the power from the prime mover to the power consumer. The electric motor, which is finding more and more extended use in the measurement of power, may be classed under this head.

144. Absorption Dynamometers. — The most common example of this form of dynamometer is the ordinary Prony brake. In Fig. 193, *ABC* is a strap or band surrounding the circumference of the wheel *W*, which is driven by the machine whose power output is to be measured. If the strap is drawn up by means of the clamp *AC* sufficiently for the friction between the strap and the surface of the wheel to overcome the weight of the strap, the latter will be carried around with the wheel unless prevented by an arm *DE* resting against a solid support at *E*. Usually in this construction of Prony brake a second arm or link *FH* is put in merely to stiffen and to help support the strap. The tendency of the strap to rotate will produce some reaction *G* at *E*, the magnitude of which depends entirely upon the friction produced by adjusting the clamp *AC*.

In the figure let a be the arm in feet at which the force G acts. Note that in all cases this arm must be measured from the center of the wheel perpendicular to the line of action of the force G (see the modifications in Fig. 194). Also let n be the number of revolutions per minute made by the wheel. Then, if the strap were free to travel with the wheel, the point E would in one minute describe a path equal to $2\,\pi an$ feet, and this path would be described against a resistance equal to G pounds. The work done therefore is $2\,\pi Gan$ foot-pounds, and the horse power developed would be

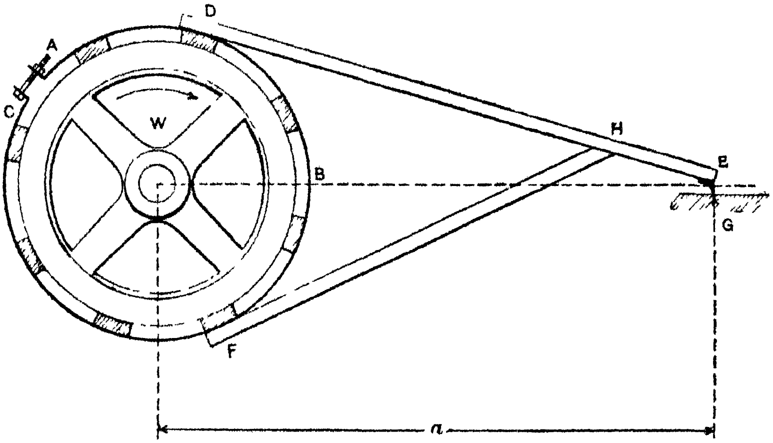

FIG. 193 —PRONY BRAKE

$$\text{horse power} = \frac{2\,\pi Gan}{33{,}000}. \qquad (6)$$

In the above application of these brakes, the factor $\frac{2\,\pi a}{33{,}000}$ in the above equation is a constant, and it becomes necessary merely to determine G and n. The latter may be found by any of the many types of speed counters. In the determination of G for the type of brake shown in Fig. 193 and in Fig. 194 A it should be noted that G is the net reaction on the scale or other apparatus used to measure it, and not the total weight shown, because the latter is too large by the action of the weight of the arm and of the support. For small and medium sized brakes, or whenever the machine whose output is measured can be slowly rotated in either direction, the simplest way of determining the *brake-zero*, as it is called, is to proceed as follows:

Let W be the weight of whatever support for the brake-arm there may be on the scale.

W_1 be the downward force due to the weight of the brake-arm.

F be the force of friction generated between the strap and the wheel when turning the wheel in either direction at the same speed.

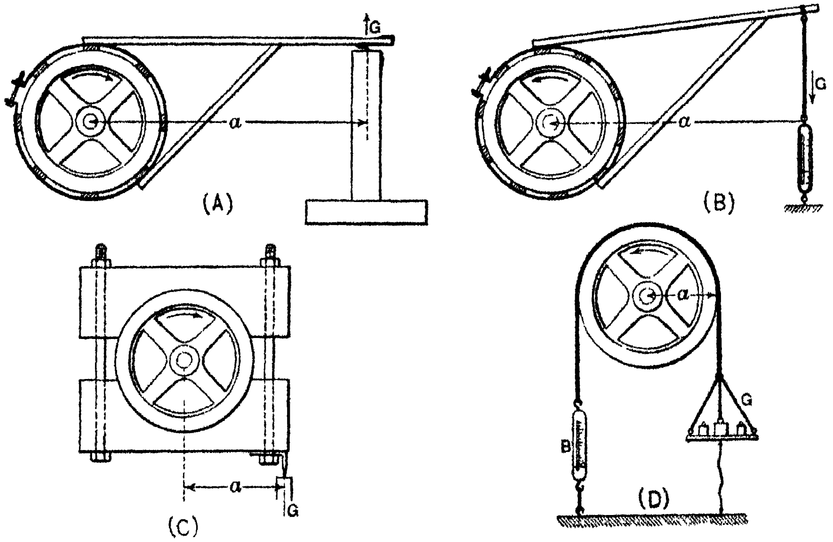

Fig. 194.—A to C. Various Types of Prony Brakes

Turn the wheel first in the direction of the arrow in Fig. 193, we have

$$G_1 = W + W_1 + F.$$

Next, turn in the opposite direction; then

$$G_2 = W + W_1 - F.$$

Therefore, by addition,

$$G_1 + G_2 = 2\,(W + W_1)$$

and

$$W + W_1 = \frac{G_1 + G_2}{2}. \qquad (7)$$

W_1 can now be obtained by itself by determining the actual weight W of the support. In most cases, however, it is not necessary to know this, and $W + W_1$ is usually known as the brake-zero reading. This must be subtracted in all cases from the total scale reading to obtain G for formula (6).

In case it should not be possible to turn the machine over slowly in either direction, as above indicated, the following procedure may sometimes be used. Place a man on the scale and let him slowly raise and depress the brake-arm through a small angle above and below its normal position and against a small friction. Care should be taken to apply the pressure to the brake-arm exactly below or above the knife-edge support with which all brake-arms should be furnished near the end. Determine the two reactions on the scale. The sum of these divided by 2 will then be the weight of the man plus the partial weight of the brake-arm resting on the scale. Subtract the weight of the man and add that of the support or pedestal for the arm, if one is used. The result will be the brake-zero.

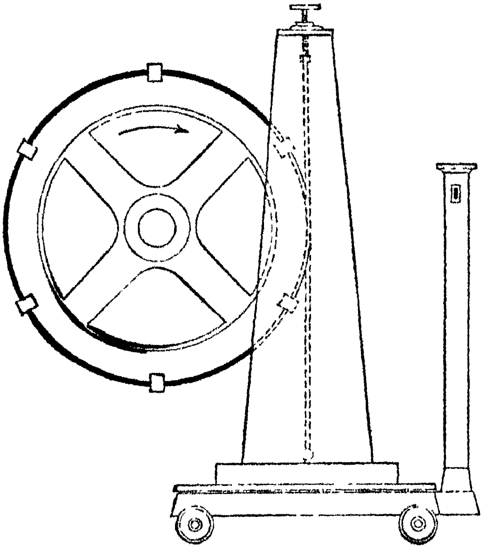

FIG. 195.—ROPE BRAKE.

In the case of the brake shown in Fig. 194 *B*, rotation as indicated by the arrow gives a reading on the balance equal to $G_1 = F - W_1$, while if the wheel is turned in the other direction, the balance must be placed above the brake-arm, and its reading will be $G_2 = F + W_1$. From this again $W_1 = \frac{G_1 + G_2}{2}$, but in this case, when the brake is used as shown in Fig. 194 *B*, W_1 must be added to, not subtracted from, the reading of the balance. In this construction $W = 0$.

In Fig. 194 *C* we have what is known as a balanced brake, i.e., $W_1 = 0$, and the brake-zero is then merely the weight W of whatever support there may be on the scale.

Fig. 194 *D* shows a very simple form of strap or band brake in which a weight *G* is placed in a scale box or pan, which weight is held floating off the floor by the friction acting on the strap laid over the pulley, as shown. The left-hand end of the strap is usually also fastened to the floor, but except as a matter of safety or convenience this is not necessary. It merely helps to guide the strap and to prevent the weights from crashing to the floor should the wheel stop. If the left-hand end is secured, care must be taken to see that the friction alone is able to overcome the weight *G*, that is, that there is no tension in the left side of the strap. To make sure of this a spring balance *B* is sometimes interposed as shown. The active force acting in this brake is evidently the weight *G* + the weight of the pan or box, while the brake arm is equal to the diameter of the wheel + ½ the thickness of strap or rope. The scale pan should be fastened by means of a stout wire, as shown, to prevent the weight being pulled over the wheel by accident.

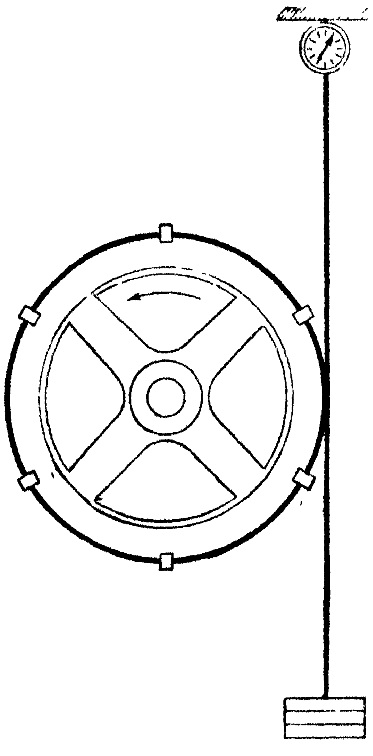

FIG. 196.—BAND OR ROPE BRAKE.

The brakes shown in Figs. 195 and 196, usually constructed as rope brakes, are modifications of the fundamental type of Fig. 194 *D*.

145. The Design of Prony Brakes. — The material for the brake may be anything that is sufficiently flexible and strong. For small brakes, leather belting, either directly in contact with the surface of the wheel or serving as a holder for a number of narrow wood cleats which are in contact with the face of the wheel, is extensively used. For medium power brakes, rope which is held in place by a number of wooden retainers, see Fig. 197, often takes the place of leather. For very heavy work, sheet steel may be employed. In such cases the contact pieces are made of wood and fastened to the inside of the steel band, or, as is done in the case of the 150-H.P. Sibley College Corliss engine, there is first a sheet-steel lining band which is in contact with the face of the wheel and takes all the wear. This is next surrounded by a copper water jacket through which water is circulated to keep the friction strap cool, and the latter is in turn held to the wheel by the sheet steel band which carries the brake-arm, drawing up-clamps, and acts as the brake-strap proper. There are two or three essential points to consider in the design of Prony brakes,—the stresses in the strap, the method of keeping the brake cool, and the method of lubricating.

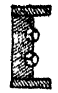

FIG. 197.—METHOD OF HOLDING ROPE.

The *stresses* in the brake-strap may be computed as follows: Let T_1 represent the greatest tension, T_2 the least tension, c the percentage that the arc of contact bears to the whole circumference, N the normal pressure, F the resistance of the brake, and f the coefficient of friction. Then

$$T_1 - T_2 = F \quad \text{and} \quad N = F \div f.$$

Now it can be shown that

$$\frac{T_1}{T_2} = 10^{2.7288fc} = \text{Number whose log is } 2.7288fc = B.$$

From which

$$T_1 = \frac{FB}{B - 1}. \tag{8}$$

and

$$T_2 = \frac{F}{B - 1}. \tag{9}$$

The actual process of designing a brake* is as follows: There are given the power to be absorbed, number of revolutions, diameter and face of the brake-wheel. In case a special brake-wheel is to be designed, the area of bearing surface is to be taken so that the number obtained by multiplying the width w of the brake in inches by the velocity v of the periphery of the wheel in feet per minute, divided by the horse power H, shall not exceed 500 to 1000. Call this result K. Then

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* See "Engine and Boiler Trials," by R. H. Thurston, pages 260 to 282; also "Friction and Lubrication."

$$K = \frac{wv}{H}. \tag{10}$$

400 to 500 is considered a good average value of K.

The value of the coefficient of friction f should be taken as the lowest value for the surfaces in contact (see table of coefficient of friction in Appendix). This coefficient is about 0.2 for wood or leather on metal, and about 0.15 for metal on metal.

Let H be the work to be transmitted in horse power, n the number of revolutions of the brake-wheel, D its diameter. Then the resistance F of the brake must be

$$F = \frac{33{,}000\ H}{\pi Dn}. \tag{11}$$

The arc of contact is known or assumed, and may be expressed as convenient in circular measure θ, degrees α, or in percentage of the whole circumference c.

Example. — Assume the arc of contact as 180 degrees ($c = 0.5$), the diameter of brake-wheel 4 feet, coefficient of friction ($f = 0.15$), face of brake-wheel 10 inches, revolutions 90, horse power 70. Find the safe dimensions of the brake-strap and working parts of the brake.

From page 281

$$B = 10^{2\ 7288fc} = 10^{0\ 2046}.$$

That is, B equals the number whose logarithm is 0.2046; or,

$$B = 1.602.$$

Since the brake-wheel is 4 feet in diameter and revolves at 90 revolutions per minute, we get from Eq. (11)

$$F = \frac{(33{,}000)\ (70)}{(\pi)\ (4)\ (90)} = 2043 \text{ pounds.}$$

Taking B as above, and substituting in equations (8) and (9), we have

$$T_1 = 2043\left(\frac{1.602}{.602}\right) = 5436;$$

$$T_2 = \frac{2043}{.602} = 3395;$$

$$N = \frac{2043}{.15} = 13{,}620.$$

From the value of T_1, the maximum tension, we next compute the required area of the brake-straps, assuming a safe working stress for the material used.

If the latter is steel, we may assume a stress of 10,000 lbs. per sq. inch, in which case

Section of brake-straps = $5436 \div 10,000 = 0.55$ square inch.

The assumed width of brake-wheel is 10 inches; this gives for the value of K, by equation (10), page 282.

$$K = (10)\ (1132) \div 70 = 162; \text{ a low value.}$$

If it is proposed in this brake to use 3 straps, each 2 inches wide, the thickness will be

$$0.55 \div 6 = 0.091 \text{ inch.}$$

146. Cooling and Lubricating a Prony Brake. — Water is the usual agent for keeping a Prony brake cool. It may be used either inside of the flanged rim of the brake-wheel, or in special forms of brakes it may be circulated through jackets surrounding the strap. In case the rim of the wheel is flanged on the inside, it may in some cases be sufficient simply to throw some water into the rim from time to time, producing the cooling action by evaporation, but where the power taken off is considerable it is better to arrange a method for supplying water to the rim and for scooping it out continuously, thus producing a constant circulation.

Concerning lubrication, it is better to use a constant quantity of oil or grease than to use either intermittently, because with intermittent supply the adjustment is disturbed every time the strap is oiled or greased. It should also be remembered in this connection, when supplying water for cooling purposes, that water will act as a lubricant if it is accidentally thrown under the strap, and will momentarily destroy the adjustment of the brake.

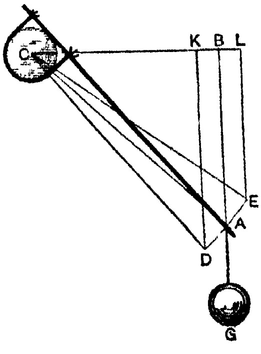


FIG. 198. — SELF-REGULATING BRAKE.

147. Other Forms of Friction Brakes: Self-regulating Brakes. — Brakes with automatic regulating devices are sometimes made; in this case the direction of motion of the wheel must be such as to lift the brake-arm. If the tension is too great the brake-arm rises a short distance, and this motion is made to operate a regulating device of some sort, lessening the tension on the brake-strap; if the tension is not great enough, the brake-beam falls, producing the opposite effect. A very simple form of *self-regulating brake*
]

is shown in Fig. 198; in this case the arm is maintained at an angle with the horizontal. If the friction becomes too great, the weight *G* rises, and the arm of the brake swings from *A* to *E*, thus increasing the lever-arm from *BC* to *LC*; if the friction diminishes, the lever-arm is correspondingly diminished, thus tending to maintain the brake in equilibrium.

Another form of self-regulating brake is shown in Fig. 199. If this brake is to be constantly loaded with the weight *P*, it is adjusted at *A* until *P* floats and until all the slack has just been taken out of the string *s*. Now if the friction should momentarily increase, the weight *P* will rise, but this causes *s* to exert a pull at *A'*. This extends the spring *S* and separates the brake levers. A decrease in the friction causes *P* to drop, which allows the spring to again exert its full force and tightens the brake.

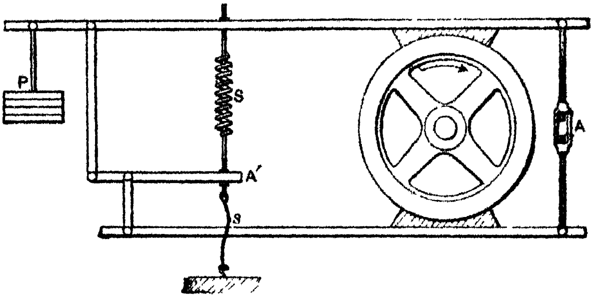
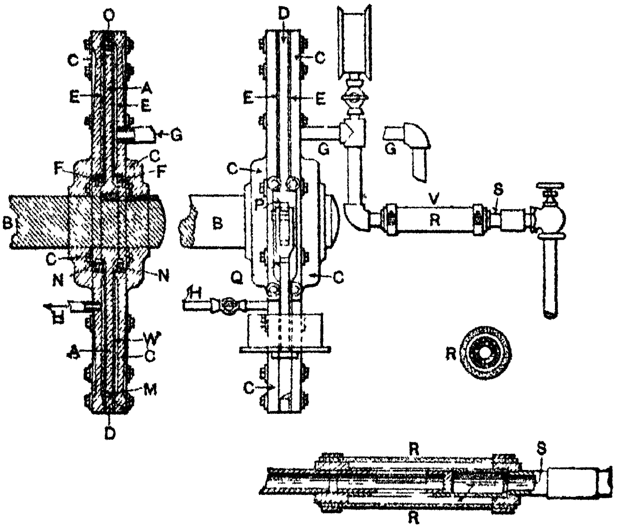


FIG. 199.—SELF-REGULATING BRAKE.

A number of various forms of automatic Prony brake may be found described in technical literature, but they are little used because in laboratory practice or for testing floors other more convenient forms, like the magnetic brake, have been invented, while in practice a single application of a brake would hardly warrant the cost of construction of a complicated form.

148. **Alden Brake.** — The Alden brake (see Figs. 200–203) is an absorption dynamometer in which the rubbing surfaces producing the friction are separated by a film of oil, and the heat is absorbed by water under pressure. It is constructed by fastening a disk of cast iron, *A*, Fig. 200, to the power-shaft; this disk revolves between two sheets of thin copper *E E*, joined at their outer edges, from which it is separated by a bath of oil. Outside the copper

sheets on either side is a chamber which is connected with the water supply at *G*. The water is received at *G* and discharged at *H*, maintaining a moderate temperature. Any pressure in the chamber causes the copper disks to press against the revolving plate, producing friction which tends to turn the copper disks. As these are rigidly connected to the outside cast-iron casing and brake-arm *P*, the turning effect can be balanced and measured the same as in the ordinary Prony brake. The pressure of water is automatically regulated by a valve *V*, Figs. 201, 202 and 203, which is partially closed if the brake-arm rises above the horizontal, and is partially opened if it falls below; with a constant head this brake gives exceedingly close regulation.



FIGS. 200–203.—ALDEN BRAKE.

Fig. 204 * shows the construction of the Alden brake more clearly than the more or less conventional sketch in Fig. 200. Here plates *A A* are the movable cast-iron disks, on each side of which are located the copper plates *B B*. The spaces between *A* and *B* are, as before, filled with oil, while the cooling water, entering through *D* and leaving through *E*, circulates through the spaces *C C*. This is one of the 400-H.P. brakes used in connection with the locomotive-testing plant at Columbia University. The method of determining the tendency to rotation of the case is shown in the conventional sketch of the lever system in Fig. 205.

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* Trans. A. S. M. E., Vol. 25, p. 861.

149. Brakes employing the Friction of Liquids. — Brakes have been constructed in which the friction of liquids has been substituted for the friction of solids. There are two distinct designs. In the

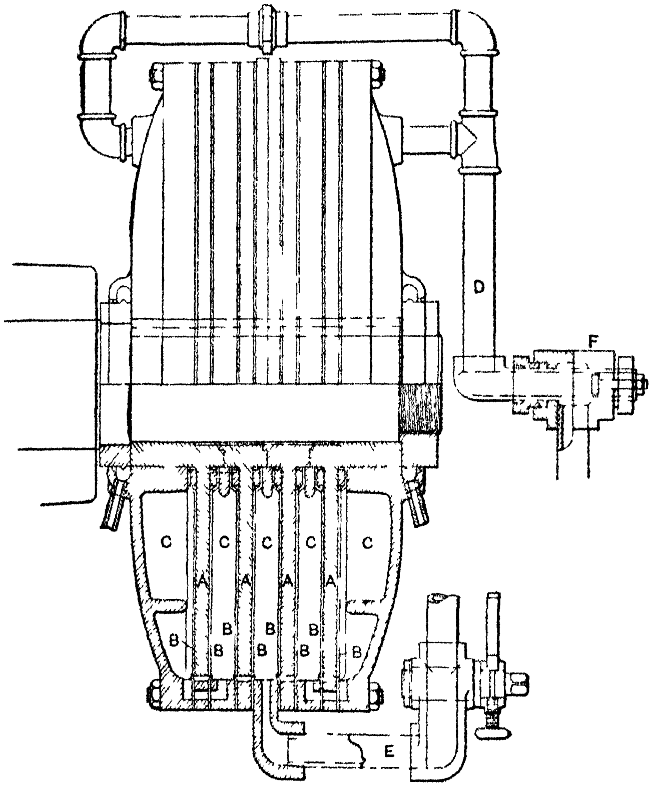


Fig. 204 — Alden Brake.

one a revolving member, notched or corrugated, disk, drum, or wheel is made to throw water against the similarly shaped inner surface of the case in which it runs. The resistance is produced by friction and impact and the power is converted into heat which may heat the water to boiling. In most cases, however, enough water may be circulated to prevent this. The outer casing is free to turn about its shaft, and the power delivered may be determined by measuring the tendency to turn just as in a Prony brake. The brake used by the Westinghouse Machine Company for testing its steam turbines is of this type. Figs. 206 and 207, reproduced from *Power*, June 30, 1908, show the general features. The rotor, provided with projecting ridges, as shown, has internal and external flanges. Water is supplied through a funnel to the interior of the rotor and is thrown outward by centrifugal force. It flows through holes bored in the projections to the outside of the rotor and strikes the ridges on the inner surface of the casing, producing resistance as above described.

Such brakes are very powerful, but subject to very rapid wear. A rotor about twenty-two inches in diameter and eight inch face will absorb about 2000 H.P. at 3600 R.P.M. The power varies approximately as the cube of the peripheral velocity. For a discussion of the wear encountered, see the above-mentioned article in *Power.*

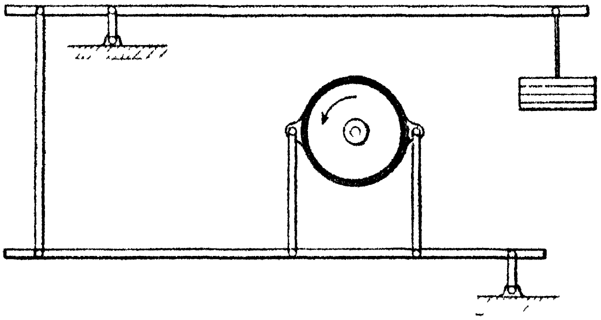
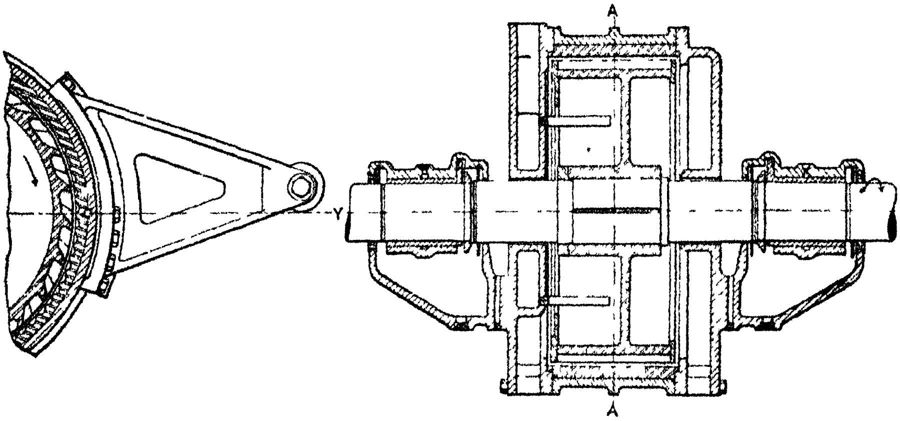


Fig. 205.—Lever System Alden Brake, Locomotive-Testing Plant, Columbia University.

The second type of liquid friction brakes is adapted only for high speeds, and differs from the former in that smooth disks, usually without any projections whatever, are used in place of corrugated or notched disks and drums. The inside of the case is usually also left smooth, although not machine finished. When water is supplied to the case and fills it up so that the lower edge of the rapidly revolving disks commences to cut through it, the water is set in motion, following the disks around, and soon will be formed into rings which travel around with the edge of the disks but at a somewhat slower rate. The friction produced between water and disk on the one hand and water and casing on the other tends to rotate the casing, which tendency is measured as before.

A brake of this type, designed by Professor Stumpf, is shown in Figs. 208 and 209.* This was used to test a 2000-H.P. turbine at from 1200 to 3400 R.P.M., a series of disks of different diameters being used. The disk shown in the picture has a diameter of 100 cm. (3.28 ft.). In other designs of this brake the number of disks was greater and the brake was supported on both sides, which would tend to quieter operation.

In either type of liquid friction brake, the power output may be easily controlled by means of regulating the water supply, and the operation of these brakes has proven very satisfactory.



FIGS. 206–207.—WESTINGHOUSE FRICTION BRAKE.

150. Other Forms of Brakes: Pump Brakes. — A rotary pump which delivers water through an orifice that can be throttled or enlarged at will has been used with success for absorbing power.

If the casing of the pump is mounted so as to be free to revolve, it can be held stationary by a weighted arm, and the absorbed power measured, as in the case of the Prony brake. If the casing of the pump is stationary, the work done can be measured by the weight of water discharged multiplied by the height due to the greatest velocity of its particles multiplied by a coefficient to be determined by trial.

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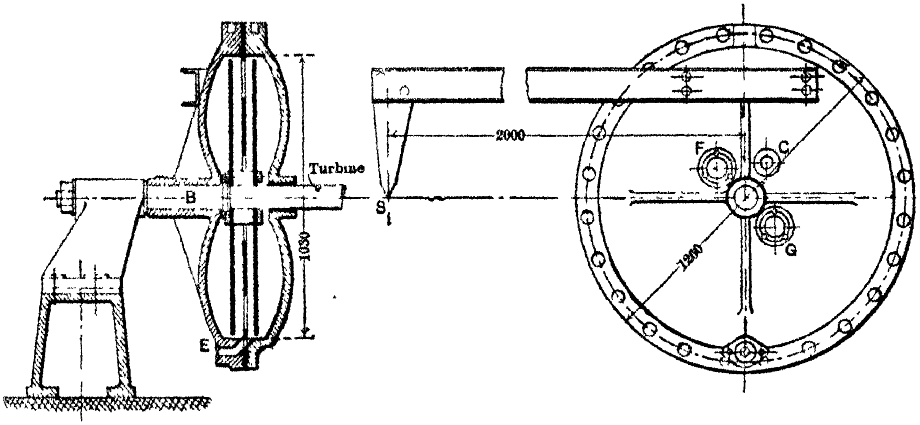
* F. Rotscher in the Zeitschrift des Vereins deutscher Ingenieure, April 20, 1907.

A special form of the pump-brake, with casing mounted so that it is free to revolve, has been used with success on the Owens College experimental engine by Osborne Reynolds. In this case the brake is practically an inverted turbine, the wheel delivering water to the guides so as to produce the maximum resistance. The water forced through the guides at one point is discharged so as to oppose the motion of the wheel at another point.

151. Fan-brakes. — A fan or wheel with vanes revolved in water, oil, or air will absorb power, and in many instances forms a convenient absorption-dynamometer.



FIGS. 208–209.—STUMPF FRICTION BRAKE.

The resistance that may be obtained from a fan-brake is expressed by the formula

$$Rl = lKDA\frac{V^2}{2g}. \tag{12}$$

in which Rl equals the moment of resistance; V the velocity in feet per second of the center of vane; A the area of the vane in square feet; l equals the distance from center of vane to axis in feet; D the weight per cubic foot of fluid in which the vane moves. K is a coefficient, found by experiment by Poncelet to have for air the value

$$K = 1.254 + \frac{1.6244\sqrt{A}}{l - s}, \tag{13}$$

in which s is the distance in feet from the center of the entire vane to the center of that half nearest the axis. When the vanes are set at an angle i with the direction of motion the value for Rl must be multiplied by

$$\frac{2 \sin^2 i}{1 + \sin^2 i}.$$

152. Electro-magnetic Brake. — If a metal disk or wheel is made to cut the field of an electro-magnet, it will experience a certain resistance which may be utilized to put machines under load. For this purpose two of these magnets are fastened to a yoke which fits over the brake wheel, as shown in Fig. 210. The resistance or drag encountered tends to revolve the yoke about the center of the wheel and its force may be measured by holding the yoke stationary. The method of computing the power is otherwise the same as for the Prony brake.

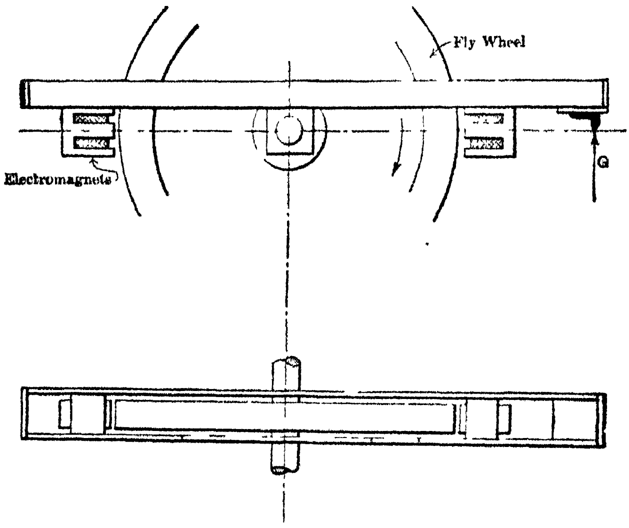


Fig. 210.—Type of Electro-magnetic Brake.

Figs. 211 and 212 show a somewhat more complicated commercial form of this type of brake.* Here a number of electro-magnets *a* are so held in an aluminum spider *c*, which is free to turn about the shaft, that the magnetic lines are forced to pass through two copper disks *d*, driven by the prime mover. The tendency of the spider to turn is measured by balancing it by means of weights along the arm *b*.

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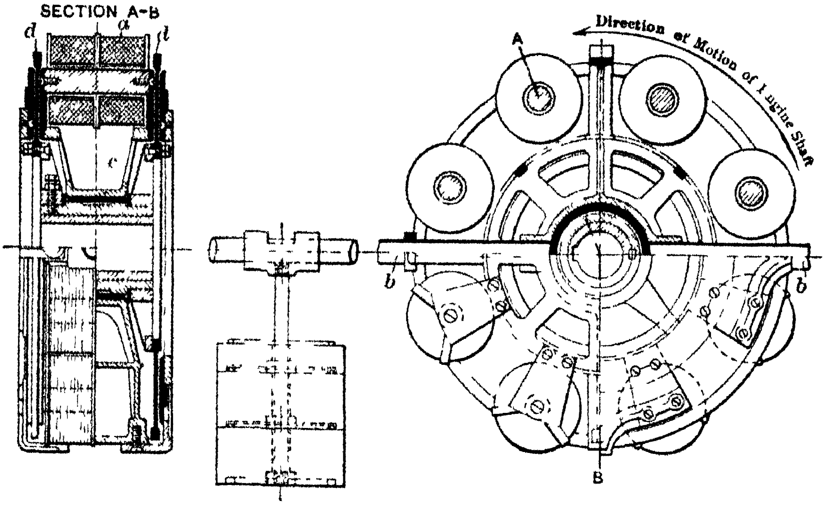
* A. Heller in the Z. d. V. D I., Oct. 5, 1907.

It will be noted that in these brakes all of the power measured is destroyed in eddy currents, appearing finally in the form of heat. The increase of temperature in the parts concerned changes the electrical resistance, lowering the capacity, and makes it necessary to stay below a certain temperature to prevent injury. There is



FIGS 211–212. — ELECTRO-MAGNETIC BRAKE.

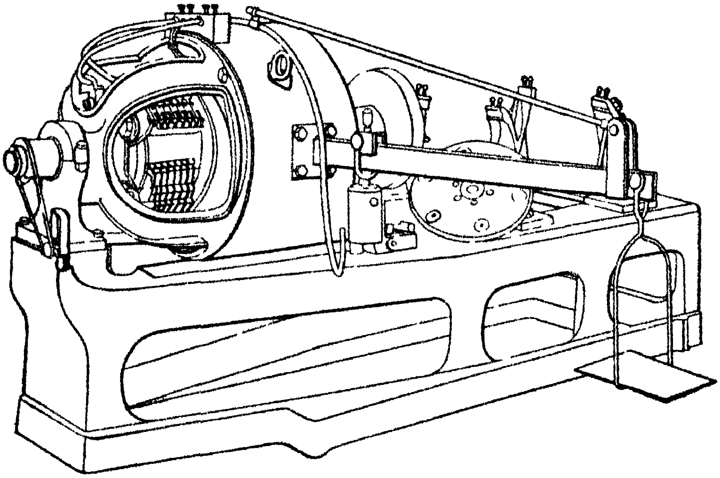


FIG. 213. — ELECTRIC GENERATOR ABSORPTION DYNAMOMETER.

no good way to keep such a brake cool. On the other hand, the power output may be easily regulated by varying the resistance in the circuit, and the first form at least may be so built as to be quickly adjusted to different sizes of flywheel.

A fundamentally similar form of magnetic brake, used where the speed variation of the prime mover is not very great, is that illustrated in Fig. 213.* Here the prime mover drives the armature of the generator, whose output is best regulated by a water rheostat. The field housing of the generator is carried by ball bearings which surround the armature bearings. The reaction between armature and field tends to turn the housing in a direction opposite to that in which the armature turns. This tendency is counterbalanced as shown. The computation of power is the same as for the Prony brake.

153. Traction Dynamometers. — *Dynamometers* for simple traction or pulling are usually constructed as in Fig. 214. Pull is applied at the two ends of the spring, which rotates a hand in proportion to the force exerted.

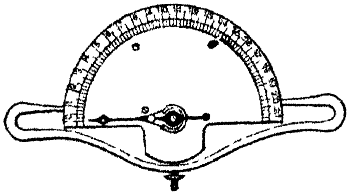


FIG. 214.— SIMPLE TRACTION DYNAMOMETER OR SPRING BALANCE.

Recording Traction Dynamometers. — These are constructed in various forms. Fig. 215 shows a simple form, designed by C. M. Giddings. Paper is placed on the reel *A*, which is operated by clockwork; a pencil is connected at *K* to the hand, and this draws a diagram, as shown in Fig. 216, the ordinates of which represent pounds of pull, the abscissæ the time. The drum may also be arranged to be operated by a wheel in contact with the ground; then the abscissa will be proportional to the space, and the area of the diagram will represent work done.

It should be noted that these instruments are not strictly dynamometers, that is, not strictly measurers of power unless by some means the velocity with which the instrument itself is bodily moved through a given distance is also recorded. This may be done by an attachment which may be an integral part of the instrument itself, but in many cases the determination of the velocity factor is accomplished entirely independently.

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* A. Heller, Z. d. V. D. I., Oct. 5, 1907.

154. Transmission Dynamometers, General Types.* — Transmission dynamometers are of different types, the object in each case being to measure the power which is received without absorbing any greater portion than is necessary to move the dynamometer. They all consist of a set of pulleys or gear wheels, so arranged that they may be placed between the prime movers and machinery to be driven, while the power that is transmitted is generally measured by the flexure of springs or by the tendency to rotate a set of gears, which may be resisted by a lever.

155. Morin's Rotation Dynamometer. — In Morin's dynamometer, which is shown in Fig. 217, the power is transmitted through springs, *FG*, which are thereby flexed an amount proportional to the load. The flexure of the springs is recorded on paper *V* by a pencil *Z* fastened to the rim of the wheel *E*. A second pencil is stationary with reference to the frame carrying the paper. The paper is made to pass under

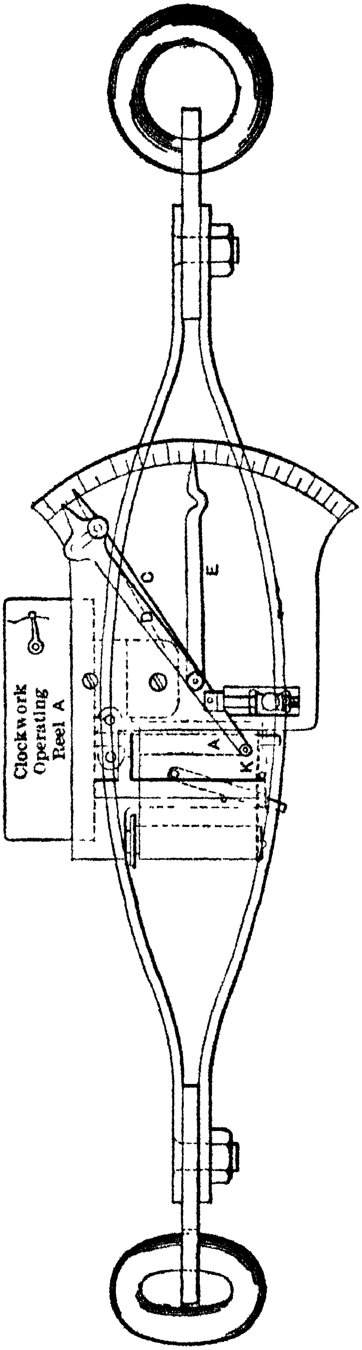

FIG. 215. — RECORDING TRACTION DYNAMOMETER.

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* See Thurston's "Engine and Boiler Trials," p. 264; also Weisbach's "Mechanics," Vol. II, pp. 39–73; also Rankine's "Steam-engine," p. 42.

the pencil by means of clockwork driven by the shafting, which can be engaged or disengaged at any instant by operating a lever. The springs are fastened at one end rigidly to the main axle, which is in communication with the prime mover, and at the other end to the rim of the pulley *E*, which otherwise is free to turn on the main shaft. The power is taken from this last pulley, and the resistance acts to bend the springs as already described. In the figure, *A* is a loose pulley, *B* is fixed to the shaft. Power is supplied through *B* and taken off through *E*.

The autographic recording apparatus of the Morin dynamometer consists essentially of a drum, which is rotated by means of a worm gear *K* cut on a sleeve, which is concentric with the main axis. This sleeve slides longitudinally on the axis, and may be engaged with or disengaged from the gear train of the frame at any instant by means of a lever. When the sleeve is engaged with the gear train, the recording apparatus is put in motion. The pencil attached to the spring will then trace a diagram on the paper whose ordinates, as measured from a base line drawn by the stationary pencil, are proportional to the force transmitted. The rate of rotation of the drums carrying the paper, with respect to the main axis, is determined in the same manner as though the gears were at rest — by finding the ratios of the radii of the respective wheels. Thus the amount of paper which passes off from one drum on to the other can be proportioned to the space passed through, so that the area of the diagram may be proportional to the work transmitted.

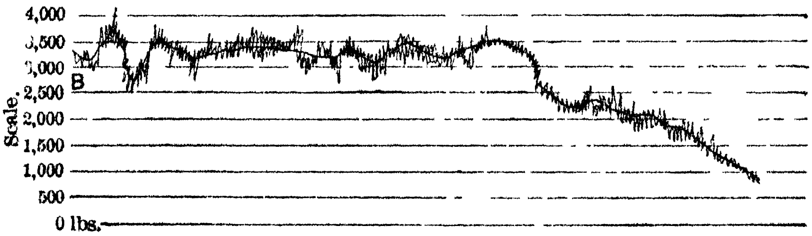

FIG. 216. — RECORD FROM DYNAMOMETER.

To find the value of the ordinates in pounds the dynamometer must be calibrated; this may be done by a dead pull of a given weight against the springs, thus obtaining the deflections for a given force; or, better, connect a Prony brake directly to the rim of the pulley *E*, make a series of runs with different loads on the brake, and find the corresponding values of the ordinates of the diagram.

Calibration of the Morin Dynamometer. — *Apparatus.* — Speed indicator, dynamometer paper, and Prony brake.

1. Fasten paper on the receiving drum, wind off enough to pass over the recording drum, and fasten the end securely to the winding drum. See that the gears for the autographic apparatus are in perfect order, and that both pencils give legible lines. Adjust the pencil fixed to the frame of the clockwork so that it will draw the same line as the movable pencil when no load is applied.

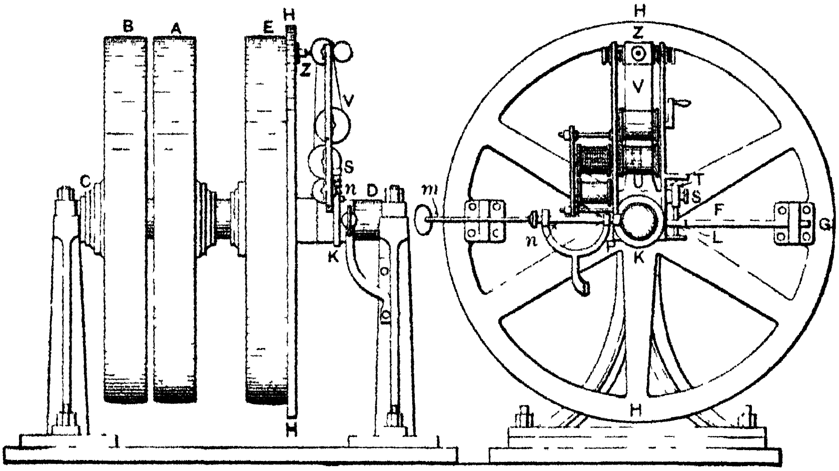

FIG. 217 MORIN DYNAMOMETER

2. With the recording apparatus out of gear apply the power. Take a diagram with no load. This will show the friction work of the dynamometer.

3. Apply power and load, take diagrams at intervals; these will represent the total work done. This, less the friction work, will be the power transmitted. The line traced by the pencil affixed to the frame of the clockwork is in all cases to be considered the zero line, or line of no work.

4. To *calibrate* the dynamometer, attach a Prony brake to the same shaft and absorb the work transmitted. This transmitted work must equal that shown by the Prony brake.

5. Draw a calibration curve, with pounds on the brake-arm reduced to an equivalent amount acting at a distance equal to the radius of the driving pulley of the dynamometer, as abscissæ, and with the ordinates of the diagrams as ordinates. Work up the equation of this curve.

6. In report of calibration make record of time, number of revolutions, brake-arm, equivalent brake-load for arm equal to radius of dynamometer pulley, length of ordinate, scale of ordinate. Describe the apparatus.

7. In using the dynamometer, insert it between the prime mover and resistance to be measured. Determine the power transmitted from the calibration.

Form of Report. —The following form is useful in calibrating this dynamometer:

CALIBRATION OF MORIN DYNAMOMETER.

Kind of brake used...... .... Length of brake-arm . ...ft.
Weight of brake-arm. .... .lbs. Zero reading of scales .. . . lbs.
Radius of driving pulley.... .ft. Observers... ....
.. ..
Date.................. .189.. .. ..

No.	Revolutions per Minute			Effective Brake-load, lbs.	Equivalent Load on Driving Pulley, lbs.	Ordinate, Inches.			Brake H. P.
	Up	Down	Mean			Up	Down	Mean.	

Remarks: Equation of Curve,
$X =$ ... . $Y =$ ....

156. Steelyard Dynamometer.—In this dynamometer, Fig. 218, the pressure Z on the axle of a revolving shaft is determined by shifting the weight G on the graduated scale-beam AC.

The power is applied at P, putting in motion the train of gear-wheels, and is delivered at Q.

Denote the applied force by P, the delivered force by Q, the radius KM by a, KE by r, LF by r_1, NL by b, the force existing at E by R, that at F by R_1.

We shall have

$$Rr = Pa, \quad \text{also} \quad R_1 r_1 = Qb.$$

But

$$R(ED) = R_1(FD);$$

and since

$$ED = FD,$$

$$R = R_1.$$

The resultant force $Z = R + R_1 = 2R$.

$$\therefore R = \tfrac{1}{2}Z; \qquad P = \tfrac{1}{2}Zr \div a; \ (14) \ Q = \tfrac{1}{2}Zr_1 \div b. \qquad (15)$$

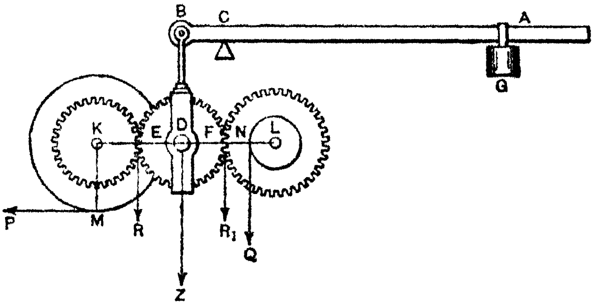

FIG. 218.—STEELYARD DYNAMOMETER.

If we know the number of revolutions, the space passed through by each force can be readily calculated, and the work found by taking the product of the force into the space passed through.

Consideration of Friction.—The friction of the axle and gear-teeth will increase the force R and decrease the force R_1. Let μ be the experimental coefficient expressing this friction. Then

$$P = \tfrac{1}{2}(1 + \mu) Zr \div a;$$

$$Q = \tfrac{1}{2}(1 - \mu) Zr_1 \div b;$$

$$\mu = \frac{Par_1 - Qbr}{Par_1 + Qbr}.$$

157. Pillow-block Dynamometer.—The pillow-block dynamometer operates on the same principle as the steelyard dynamometer, but no intermediate wheel is used. This dynamometer, shown in

Fig. 219, consists of the fixed shaft *L*, which is rotated by the power *Q* applied at *N*. The power rotates the gear-wheel *EL*, which communicates motion to the wheel *KE* on the same shaft with the wheel *KM*. This shaft is supported on a pair of weighing scales so that the force *Z* acting on the bearing can be weighed. Let *P* equal the force delivered, let α equal the angle this force makes with the horizontal, *KM* equal *a* and *KE* equal *r*, *G* equal the weight of shaft and wheel at *K*. Taking wheel *KE* as a free body and placing the summation of vertical forces = 0, we have

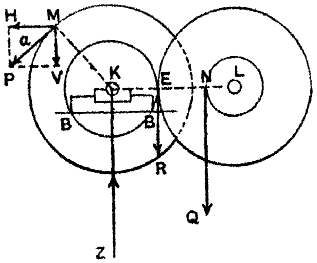

FIG. 219.—PILLOW-BLOCK DYNAMOMETER.

$$Z = G + P \sin \alpha + R = G + P \sin \alpha + \frac{a}{r} P$$

$$= G + P\left(\sin \alpha + \frac{a}{r}\right)$$

From which

$$P = \frac{Z - G}{\sin \alpha + \frac{a}{r}}. \tag{16}$$

When the belt is horizontal,

$$\alpha = 0 \quad \text{and} \quad P = (Z - G)\frac{r}{a}. \tag{17}$$

158. The Lewis Dynamometer.*—This transmission dynamometer is a modified form of the pillow-block dynamometer, arranged in such manner that the friction of the gearing or journals will not affect the reading on the weighing scales. This machine is shown in Fig. 220, and also in Fig. 239, Article 169, page 319. The dynamometer consists of two gear-wheels *A* and *C*, whose pitch-circles are tangent at *B*; the gear-wheel *A* is carried by the fixed frame *T*, the wheel *C* is carried on the lever *BD*; the lever *BD* is connected to the fixed frame *T* by a thin steel fulcrum, as used in the Emery testing-machines (Fig. 58, page 81). The point *D*, the center of wheel *C*, and the fulcrum *B* are in the same right line. The fulcrum permits vertical motion only of the point *D*. The point *D* rests on a pillar, which in turn is supported by a pair

* See Vol. VII, p. 276, Trans. Am. Society Mechanical Engineers.

of scales. The shaft leading from the wheel *C* is furnished with a universal joint (see Fig. 239), so that its weight does not affect that on the journal *C*. In Fig. 220, *A* is the driving and *C* the driven wheel, the force to be measured being received on a pulley on the shaft *a*, transmitted through the dynamometer, and delivered from a pulley on the shaft *c*. From this construction it follows that no matter how great the friction on the journals of the shaft *c*, there will be no pressure at the point *D* except what results from torsion of the shaft *c*. This will be readily seen by considering: 1. That any downward force acting at *B* will be resisted by the fixed frame *T*, and will not increase the pressure at *D*. 2. A downward force acting on the lever between *B* and *D* will produce a pressure proportional to its distance from *B*. 3. If the driven wheel *C* were firmly clamped to its support, no force acting at *B* would change the pressure at *D*; and since journal friction would have the effect of partially clamping the wheel to the journal *c*, it would have no effect on the scale reading at *D*.

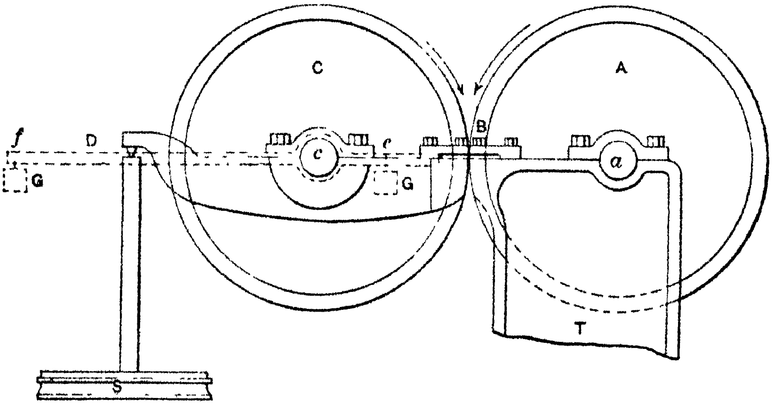

FIG. 220. LEWIS DYNAMOMETER.

Denote the transmitted torsional force by Z; the radius of the driven pulley by r; the length of lever BD by a; the scale reading at D by W. Then from equality of moments

$$Wa = Zr, \qquad Z = \frac{Wa}{r}. \tag{18}$$

The effective lever-arm BD is to be obtained experimentally as follows: Disconnect the universal joint, shown in Fig. 239, so as to leave the wheel *C* free to turn; block the driving pulley *A*; fasten a horizontal arm *ef* (dotted lines, Fig. 220) to the shaft *c*, parallel to the line *DB* and carrying a weight *G*; balance the scales in this position, then move the weight out on the lever until the reading of the scales is increased an amount equal to the weight moved. The distance moved by the weight will equal the length of the lever *DB*.

Thus let *ef*, shown in dotted lines, represent the lever clamped to the axis *c*; let *e* represent the first position of the weight *G*, and *f* the second position; let *W* and *W′* represent the corresponding weights above the reading of the scale balanced without *G* on the lever *ef*.

Then we have

$$W = G\frac{(cB)}{DB};$$

$$W + G = W' = G\frac{(fB)}{DB}.$$

Hence

$$G = \frac{W' - W}{DB} = G\frac{(fB - cB)}{DB} = G \cdot \frac{cf}{DB}.$$

Then will

$$DB = cf.$$

159. The Differential Dynamometer. — This is often called the Bachelder, Francis, or Webber dynamometer; it was invented by Samuel White, of England, in 1780, and brought to this country by Mr. Bachelder in 1836.

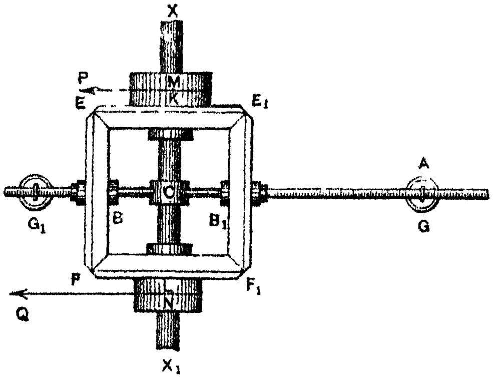

FIG. 221. — GEARING OF DIFFERENTIAL DYNAMOMETER.

The dynamometer portion consists of four bevel gears, shown in plan in Fig. 221.

Power is applied to the pulley *M*, which carries the bevel-wheel EE_1; the resistance is overcome by the pulley *N*, which carries the bevel-wheel FF_1. Both wheels run loosely upon the fixed shaft XX_1, and are connected by the wheels *EF* and E_1F_1. By the action of the force P and the resistance Q, the pressure of the wheels EE_1 and FF_1 is downward at E and F, and upward at E_1 and F_1, tending to swing the lever GG_1 around the axis XX_1 one half as fast as the speed of the pulley M. The weight which holds the lever-arm stationary, multiplied by the space it would pass through if free to move, is the measure of the work of the force P. A dashpot is usually attached to the lever GG_1 at G_1, to lessen vibrations and act as a counterbalance. Let Z equal the vertical forces acting at B and B_1; R the vertical pressure between the teeth at each point of contact; b the distance of B and B_1 from the center C; a the distance AC to the weight G.

Then we have evidently

$$2Z = 4R, \quad \text{or} \quad Z = 2R;$$

also

$$Ga = 2Zb = 4Rb.$$

If a' is the radius of the driving pulley M, and r the radius of each bevel gear,

$$Pa' = 2Rr, \quad \text{or} \quad P = \frac{2Rr}{a'} = \frac{Gra}{2ba'}. \qquad (19)$$

If friction is considered,

$$P = (1 + \mu)\frac{Gra}{2ba'}. \qquad (20)$$

The mechanical work received is equal to P multiplied by the space passed through by the point of application of P in the given time.

This instrument has been improved by Mr. S. Webber, as shown in Fig. 222.

These dynamometers are used in substantially the same way as the Morin dynamometers.

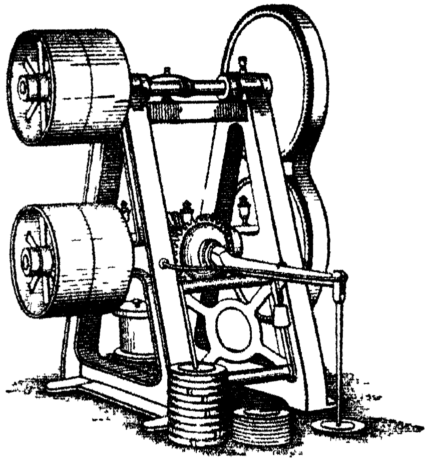

FIG. 222.—WEBBER DYNAMOMETER.

Calibration of Webber Dynamometer.—1. See that the machine is well oiled and otherwise in good condition.

2. Obtain the constants of the machine and apparatus used:
   (*a*) The brake constants (arm length and brake zero).
   (*b*) The actual weight of the sliding poise and of each of the 1000 and 2500 ft.-lb. weights belonging to the machine.
   (*c*) Measure the arms from the center of the shaft to the first notch on the beam, to the last notch, and to the knife-edge of the stationary poise.
   (*d*) Place the platform scale (used as the brake scale during the runs) under the stationary poise and put the sliding poise in the zero notch. Operate the dynamometer forward and then backward by hand at the same rate of speed, and determine the reactions on the scale. Divide the sum by 2 and the result is the unbalanced weight of the dynamometer beam concentrated at the knife-edge.

3. From the observations under 2 compute the value in foot pounds per 100 revolutions of the driving pulley for each kind of weight, for the sliding poise in the first and in the last notch on the beam, and for the unbalanced weight of the beam. (Remember that while the dynamometer pulley is making 100 revolutions the dynamometer beam would make only 50.) From the foot-pound values compute the "dynamometer reading from machine constants" (see form below).

4. Put the machine in operation with the brake off and balance the beam. This reading is called the "zero reading by beam" (= W_0 in form below).

5. Put brake in place, start the machine, and adjust the first load. Observe time of 100 revolutions, brake load, reading of balanced dynamometer beam, and note the number and kind of weights used together with the position of the sliding poise. Make the same observations for a series of loads up to the capacity of the dynamometer.

6. From the observed quantities fill out the form below.

7. Draw a calibration curve between ft.-lbs. per 100 revolutions as read off the dynamometer beam and as measured at the brake.

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Calibration of.......................................Differential Dynamometer

Kind of Brake used. ....

Length of Brake Arm.............ft. Weight of Brake Arm...............lbs.

Zero reading of Brake Scales.......lbs.

Date.... ....19....

Observers
....
....
....

Number.	Time in Seconds of 100 Revolutions.	Brake Load, lbs.		Work in ft.-lbs. per 100 Revolutions					Brake Horse Power
					Dynamometer Readings.				
		Gross.	Net.	Obtained from Brake.	Observed on Dynamometer Beam.	Calculated from Machine Constants	Transmitted as shown by Beam, $= W_t$	Error of the Dynamometer.	
				W_b	W_d	W_e	$W_d - W_0$	$W_t - W_b$	D.H.P
1									
2									
3									
4									
5									
6									
7									
8									
9									
10									

MACHINE CONSTANTS.

Loads at Knife-edge	Moment Arm. ft		Data for Beam.	Sliding Poise, Weight... lbs	
	Weight, lbs.	Value ft.-lbs. per 100 Revs		Moment Arm, Feet	Value, ft. lbs. per 100 Revs.
1000 Weight			First Notch		
2500 Weight......			Last Notch.....		
Dynamometer Beam.		... $= W_e$	Increase per Notch		

$W_0 =$ Zero Reading by Beam...... ft.-lbs.

160. Emerson's Power Scale. — One of the most complete transmission dynamometers is shown in Fig. 223, with attached numbers showing the dimensions of the various sizes manufactured. In this instrument the wheel *C* is keyed or fastened to the shaft; the wheel *B* is connected with the wheel *C* near its outer circumference by projecting studs; the amount of pressure on these studs is conveyed by bent levers *KA* to a collar sliding on the shaft, which in turn is connected with weighing levers. Small weights are read off from the scale *D*, and larger ones by the weights in the scale pan *N*. A dashpot *S* is used to prevent sudden fluctuations of the weighing lever.

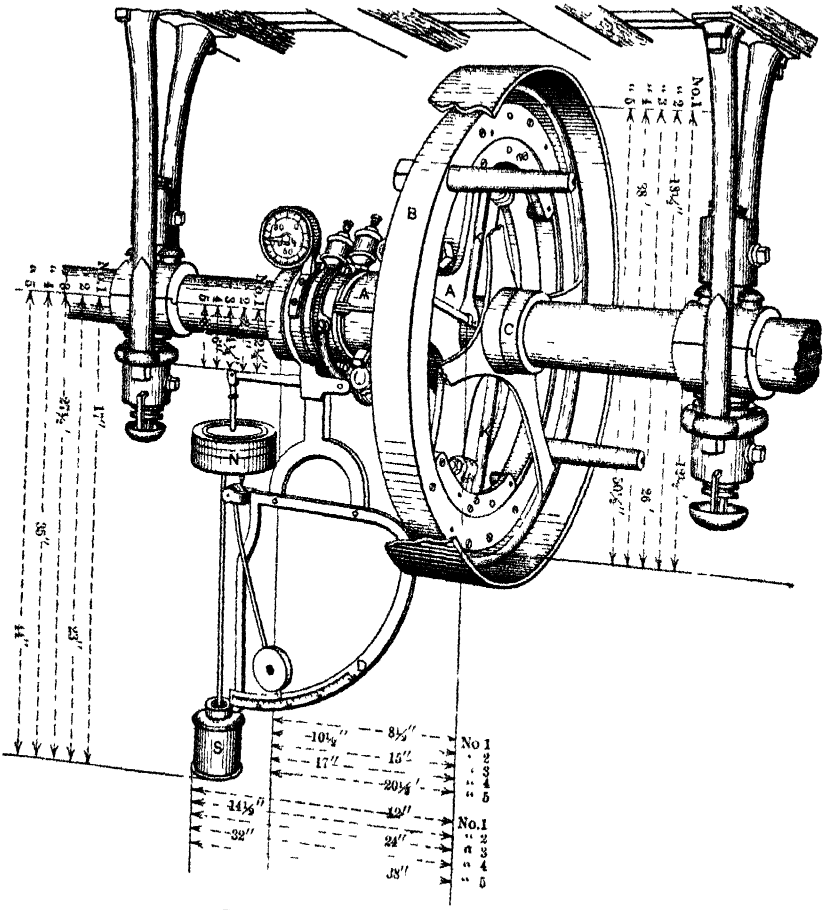

FIG. 223.—EMERSON POWER SCALE.

The apparatus can be easily calibrated by mounting a Prony brake wheel on the shaft and comparing the power output readings obtained with the scale readings of the instrument; or a known torque may be applied to pulley *C* while the shaft is held stationary.

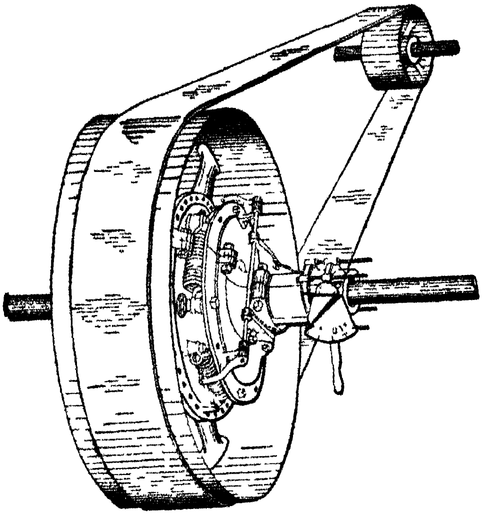

FIG. 224.—VAN WINKLE POWER METER.

161. The Van Winkle Power Meter.—The Van Winkle power meter is shown complete in Fig. 224, and with its parts separated in Fig. 225. It consists of a sleeve with attached plate, *B*, that can be fastened rigidly to the shaft, and a plate *A*, which is revolved by the force communicated through the springs *SS*. The angular motion of the plate *A* with reference to *B* will vary with the force transmitted. This angular motion is utilized to operate levers and move a loose sleeve longitudinally on the shaft. The amount of motion of the sleeve, which is proportional to the force transmitted, is indicated by a hand moving over a graduated dial. The dial is graduated to show horse power per 100 revolutions.

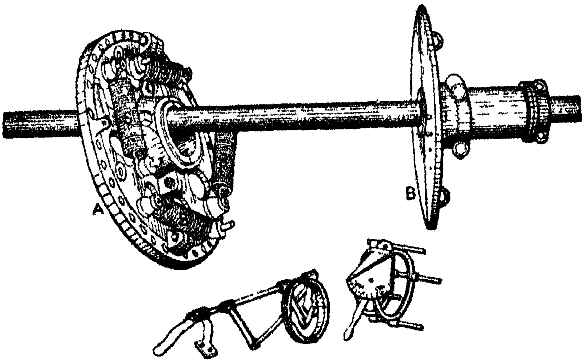

FIG. 225 — PARTS OF VAN WINKLE METER.

162. Belt Dynamometers. — Belts have been used in some instances instead of gearing in transmission dynamometers, but because of the great loss of power due to stiffness of the belts, and on account of the uncertainty caused by slipping, they have not been extensively used. The following form, from Church's "Mechanics of Materials," is probably as successful as any that has been devised. It consists of a vertical plate, carrying four pulleys and a scale pan, as shown in Fig. 226. The scale beam is balanced, the belt then adjusted, and power turned on; a sufficient weight, *G*, is placed in the scale pan to balance the plate again. Let *b* be the arm of the scale pan, and *a* that of the forces *P* and *P'*. Then, for equilibrium,

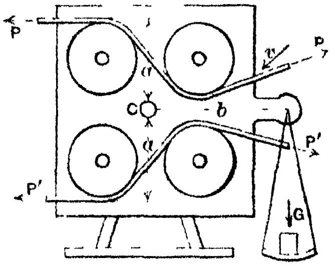

Fig. 226 —Belt Transmission Dynamometer.

$$Gb = Pa - P'a, \tag{20}$$

since *P* and *P'* on the right have no leverage about *C*, as the line of the belts produced is made to intersect *C*. From (20)

$$P - P' = \frac{Gb}{a}. \tag{21}$$

The work transmitted in foot-pounds per minute is equal to $(P - P')v$, in which *v* is the velocity of the belt in feet per minute to be obtained by counting. Another form employs two quarter-twist belts to revolve a shaft at right angles to the main shaft. (See Vol. XII, Transactions Am. Soc. Mechanical Engineers.)

163. The Durand Dynamometer.—This is a special form of belt dynamometer designed by Prof. W. F. Durand and used by him in screw propeller investigations. It is described in Vol. XXVIII of the Transactions of the American Society of Mechanical Engineers. The following description and the illustrations are reproduced here from that article:

A and *B*, Fig. 227, are two sprocket wheels mounted on a frame *XY* which is carried at *E* by a steel spring support, after the manner of the well-known Emery steel-plate knife-edge. This is frictionless, and for small movements of the frame the bending stress developed is infinitely small in comparison with the other forces in play. As shown at *SS*, the frame carries semicircular plates of steel fitted with a slot, either end of which engages with a pin as the frame oscillates through its permitted range of motion of about one-fourth inch at a distance of ten inches from the center of support. This permits perfect freedom of motion within this range, but prevents movement beyond to such an extent as to develop any sensible forces within the steel spring supports.

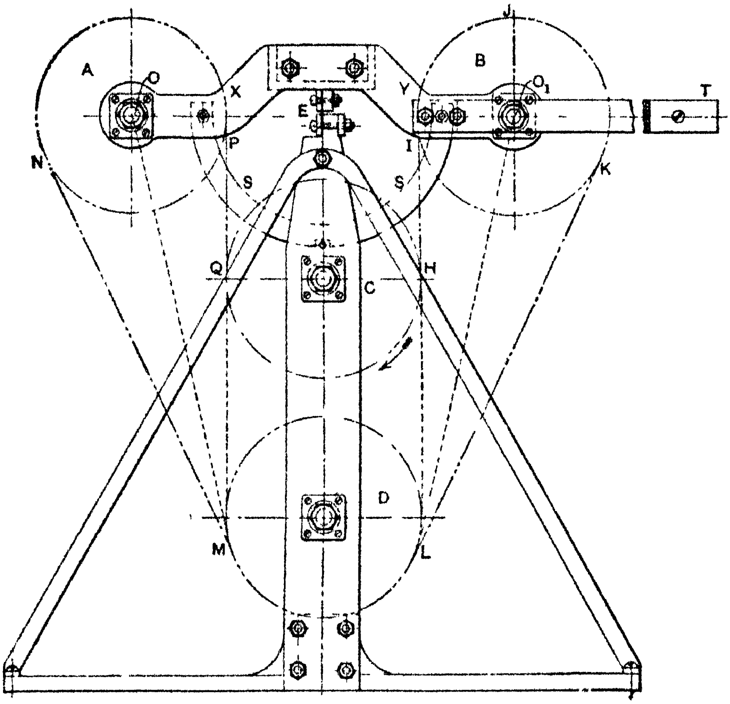

FIG. 227.—DURAND DYNAMOMETER.

The sprockets *C* and *D* are similar in form and size to *A* and *B* and are mounted on standards attached to the base. All sprockets are mounted on ball bearings in order to reduce friction to the minimum. Around these wheels is led a chain, the dimensions being so adjusted that with the proper length of spring for bearing at *E* the chain throughout its length will run with the proper amount of slack for smooth and steady operation.

The similarity between this form of dynamometer and the Tatham form will be readily noted. The present type may, in effect, be considered as a development of the six-equal-wheel form by the elimination of the two lower wheels and the substitution of chain for belt drive.

The shafts which carry the sprockets *C* and *D* are extended, carrying pulley wheels at *F* and *G*, Fig. 228, and also universal joint couplings for direct connection to a driver or follower. The power can be put in at *C* and taken out at *D* or *vice versa*, and on either side of the dynamometer as desired. If the power passes through from *C* to *D*, and *C* turns as indicated by the arrow, it is readily seen that the tight side of the chain will be from *H* around through *IJKLM*, and the loose side from *M* through *NP* to *Q*. Thus *HI* and *KL* will be under the higher or driving tension T_1, equal on both sides except for the small friction of the ball bearing on which *B* turns. Likewise, *MN* and *PQ* will be under the loose or following tension T_2, the same on both sides except for the friction of the ball bearing on which *A* runs.

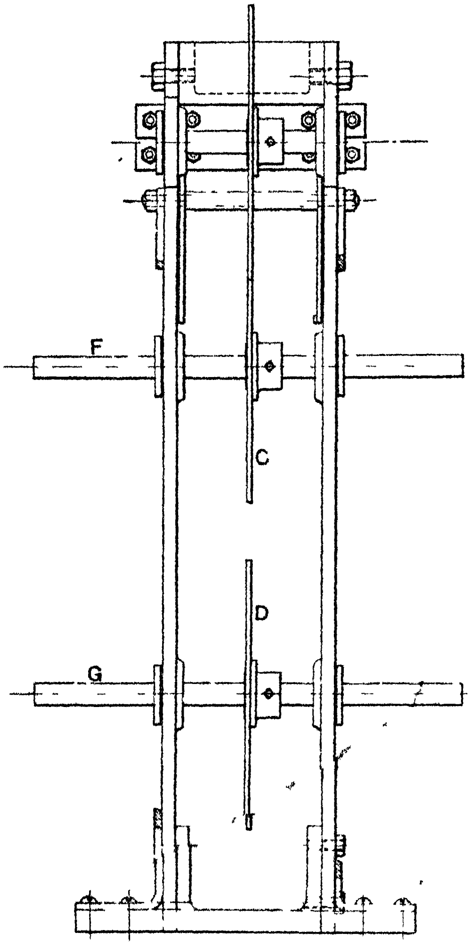

FIG. 228. — DURAND DYNAMOMETER

Disregarding these small frictional resistances, the two tensions T_1 will have the resultant $2\,T_1 \cos \theta$ directed along the line O_1L, while the two tensions T_2 will have the resultant $2T_2 \cos \theta$ directed along OM. The components of these forces perpendicular to the line OO_1 will be in each case $2\ T_1 \text{ Cos}^2 \theta$ and $2\ T_2 \text{ Cos}^2 \theta$. Denoting the distance O_1E by a_1 we have for the net turning moment about E

$$M_1 = 2\ a\ (T_1 - T_2) \text{ Cos}^2 \theta.$$

Again denoting by r the pitch-line radius of the sprockets, we have for the power taken off at D, the turning moment,

$$M_2 = r\ (T_1 - T_2).$$

Hence
$$M_2 = \frac{M_1 r}{2\ a \text{ Cos}^2 \theta}$$

and
$$\text{work per min.} = 2\ \pi n M_2 = \frac{\pi n M_1 r}{a \text{ Cos}^2 \theta}. \qquad (22)$$

Obviously the two tensions T_1 on the right, acting against the two tensions T_2 on the left, will determine a moment tending to depress the end T of a lever attached to the frame XY. If we denote ET by h and the force at T required to balance the moment by F, we shall have

$$M_1 = Fh,$$

or
$$\text{work per min.} = \frac{\pi n F r h}{a \text{ Cos}^2 \theta}. \qquad (23)$$

By careful measurement of the dimensions of the dynamometer the relation between the power transmitted and the observed values of n and F can be determined. As in all cases involving the use of such apparatus, however, it will be more satisfactory to calibrate directly by mounting a friction brake on a belt wheel attached to the delivery shaft and noting the relation between brake readings and the values of the force F. Such a calibration will, of course, serve to eliminate all friction between the points of intake and delivery, or in the dynamometer itself, and will thus serve to relate correctly the force records as measured by F with the actual power delivered.

164. Torsion Dynamometers. — When power is being transmitted through a shaft, the latter will undergo a certain twisting action, the magnitude of the total angle of torsion depending upon the amount of power transmitted, the diameter and length of the shaft, and the quality of the material. Where the amount of power transmitted is considerable, as in the case of marine engines, dynamometric measurements on the shaft itself have often been made to determine the developed horse power it carries. In such cases the measurement reduces itself to the determination of the angle of torsion and of the rotative speed. On the other hand, when the power transmitted is smaller, as in ordinary line shafting, the shaft is often broken, and springs or other more or less flexible members are then inserted, connecting the ends of the shafting, and from the relative angular displacement of the two shafts the power transmitted may then be computed. Of this type are the Morin dynamometer, described in Art. 155, while the Emerson power scale and the Van Winkle power meter operate in a similar way.

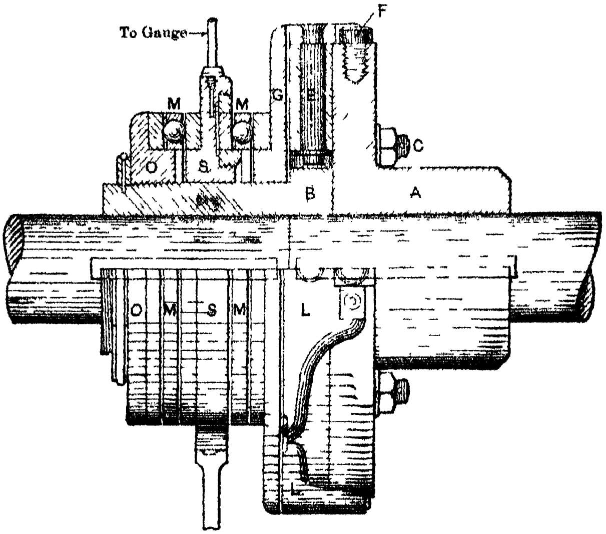

FIG. 229.—KENERSON TORSION DYNAMOMETER.

The measurement of the angular displacement may be made in a variety of ways, although the method of determining it by finding the relative displacement of two commutator segments, one on each shaft, by electrical means, is perhaps most often used. A true torsion dynamometer is the Kenerson, described in the Journal of the Am. Soc. of Mech. Eng'rs for May, 1909.

Figs. 229, 230, and 231 * show its construction. *A* and *B* are two flange couplings rigidly fastened to their shafts, but only loosely connected to each other by stud bolts *C*. The holes for *C* in *A* are so large that bolts *C* take no part in the transmitting power. *B* carries four latches *L* which pivot freely about *E*. Projecting fingers on each latch *L* surround studs *F* fastened to the circumference of *A*. It must be evident that no matter whether *A* or *B* is the driver, any attempt to transmit power will turn the latches *L* about *E*. This movement, however, is resisted by knife-edges

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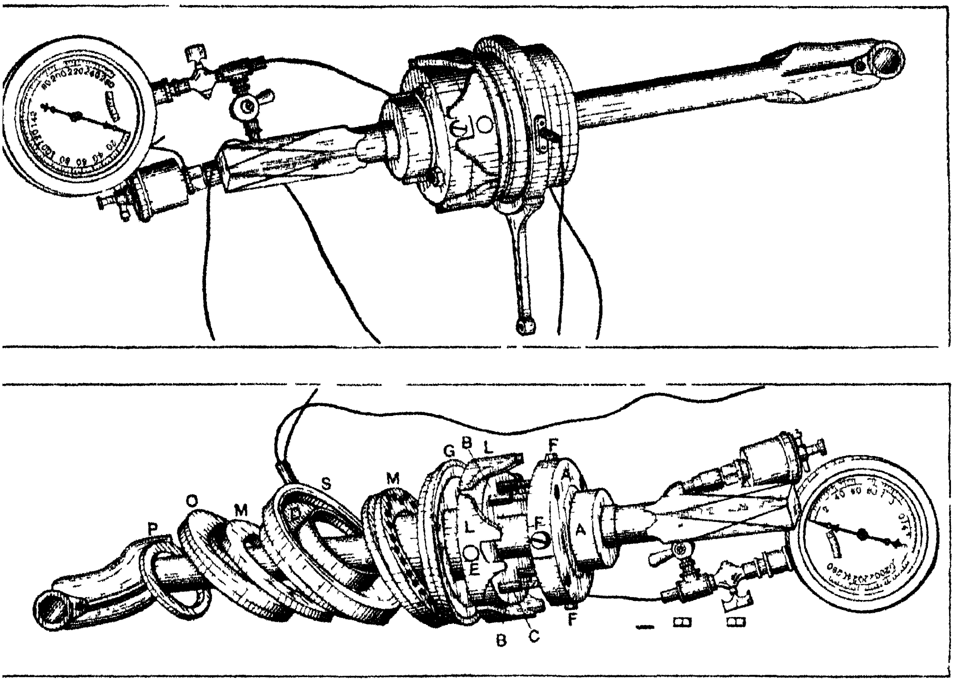
* Reproduced from the journal mentioned.



Figs. 230 231 Kenerson Torsion Dynamometer

on *L* coming in contact with the pressure plate *G*, which action tends to force the latter to the left. The pressure thus exerted is a measure of the power transmitted, and the problem then reduces to the determination of this axial thrust. The stationary ring *S* is on one side held against the ball thrust bearing *O*, while it receives the thrust of *G* through another ball bearing on the other side. In this ring *S* is cut an annular cavity covered by a thin flexible diaphragm *D*, Fig. 231. Against this diaphragm the slightly chamfered edge of the ball race *M* presses, thus transferring the thrust to the liquid, oil in this case, contained in the ring cavity. The pressure may then be measured by Bourdon gauge or other means. The instrument is very easily calibrated by comparing gauge pressures with moments applied to the flange *A*. The construction is very compact. The movement of the diaphragm is very small, so that at equilibrium of the gauge there is no fluid friction. The gauge readings of course must in any case be corrected for static head if the gauge is placed above or below the coupling itself.

The interest in torsion meters applied to solid shafting has of late years been revived on account of the desire to determine the power developed by steam turbines. The latter cannot be indicated like a reciprocating steam engine. Torsion meters for this service may be classified as electrical or mechanical, depending upon the method used to record the angle of torsion. In the case of a steam turbine operating against a constant load it is usually sufficient to determine the torsion angle θ at one point in the circumference of the shaft. It has been assumed that this could be done also for marine steam turbines in making torsion measurements, but Föttinger* has probably conclusively shown that the assumption is not justified on account of the varying resistance offered by the propeller. For that reason, measurements at several points around a circumference must be made to determine an average value of θ, just as must be done in the case of reciprocating engines, whether on land or sea.

If the average angle of torsion θ be expressed in degrees, the horse power transmitted by a solid shaft will be

$$\text{H.P.} = \frac{\theta d_e^4 n}{CL} \tag{24}$$

and for a hollow shaft

$$\text{H.P.} = \frac{\theta(d_e^4 - d_i^4)n}{CL}, \tag{25}$$

where d_e = external diameter of the shaft, in inches.
d_i = internal diameter of hollow shaft, in inches.
n = revolutions per min.
L = length of shaft, in inches.
C = 3.27 (taking the modulus of rigidity = 11,250,000).

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* Zeitschrift des Vereins deutscher Ingenieure, June 6, 1908.

If the shaft has couplings, subtract their axial length in determining L, on the assumption that they will not twist.

As an example of the electrical method of measuring θ, the Denny and Johnson torsion meter* may be cited. Fig. 232 gives a conventional sketch illustrating the principle upon which this instrument operates. For a detailed description of the latest form the reader is referred to the article cited below. Two disks *a* and *b* are fastened to the shaft so that the distance between them is as large as possible. Each disk carries a permanent, chisel-pointed magnet, indicated by *c* and *d*. Once every turn these magnets pass over electromagnets *e* and *f*, inducing a current in their windings. If these current impulses are simultaneous, they will neutralize each other and no sound will be heard in the telephone receiver *g*. When the shaft is revolving under no load, the two magnets will pass over the point *e* and the adjustable point *f*, so that a straight line joining these points is parallel to the axis of the shaft, that is, the shaft is under no torque, except the inappreciable amount of torsion due to friction in the bearings. If now the shaft is put under load, the relative positions of *c* and *d* shift; the current impulses are no longer simultaneous and distinct clicks may be heard in *g*. By turning *h*, thus shifting the point *f*, the sounds may again be made to neutralize each other, and the amount that *f* had to be moved by the micrometer screw to reproduce this condition is a measure of the angle of torque. Later forms of this instrument are so modified that both the receiver and the indicator may be placed in any quiet place at considerable distances from the shaft under investigation, and provision is also made that several sets of magnets may be placed around the circumference of the shaft, thus determining the variation of torque.

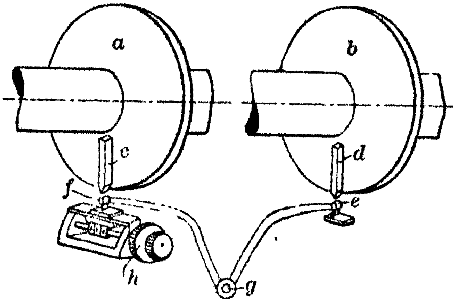

FIG. 232.—PRINCIPLE OF DENNY AND JOHNSON TORSION METER.

An entirely mechanical form of torsion meter, which however takes a continuous record of the variation of the angle of torsion, is the Föttinger torsion meter,* shown in the conventional sketch, Fig. 233. In this construction a disk, marked No. 1, is carried by a piece of tubing, which is concentric with the shaft and fastened rigidly to the shaft at the opposite end. Another disk, No. 2, is fastened in close proximity to the first disk. Any torsion in the shaft will cause a relative displacement of the two disks, and this displacement is communicated to an arm carrying a pencil by the linkage shown. The pencil makes a record on a drum, which consists of a piece of tubing surrounding the shaft and which may be moved in an axial direction along a guide. The test length is indicated in the figure. This length is limited in the first place by the distance between shaft bearings and in the second place by any possible vibration or torsion which may be set up in the tube carrying disk No. 1.

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* Transactions of the Institution of Naval Architects, 1907.

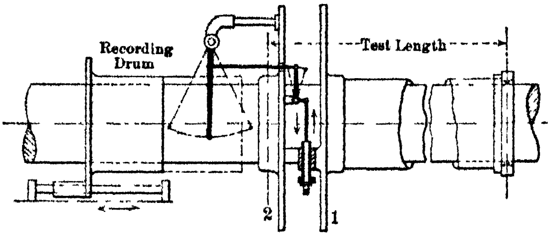

FIG. 233. — FÖTTINGER TORSION METER.

Torsion meters in which a beam of light is used to indicate the amount of torsion have been designed by Hopkinson and Thring † and by Bevis-Gibson.‡

165. Cradle Dynamometers. — These have found extended laboratory use in the testing of small dynamos and motors. In Fig. 234 the machine to be tested is rigidly secured to the cradle *A*, which at each side rests by means of knife-edges *B* upon the floor-stands *C*. The axis of the armature is adjusted exactly in line with the knife-edges. With the machine standing still, the cradle is balanced by means of the balance weight *E* and the poises *F* and *F'*. When set in motion the magnetic drag set up between armature and pole pieces unbalances the cradle. From the moments required to re store balance the power output or input may then be computed.

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* E. M. Speakman, Inst. of Eng. and Shipbuilders of Scotland, Vol. L.

† London Engineering, June 14, 1907.

‡ London Engineering, Feb. 7, 1908.

166. Electrical Measurement of Power Input and Output. — Dynamos and motors form very convenient dynamometers for determining output and input, and the method of connecting them up and taking the observations will be briefly considered.[*]

We may have two possible cases. If a power consumer is operated by a motor, we may take readings of the electrical input and multiplying this by the efficiency of the motor at the load used, we will have the power delivered to the consumer. If on the other hand a generator is used as a brake for a prime mover, the electrical output must be divided by the generator efficiency at that load to obtain the power delivered by the prime mover to the generator. In the last case, unless the current can be usefully employed, it also becomes necessary to furnish means for destroying the electrical energy, *i.e.*, for loading the generator.

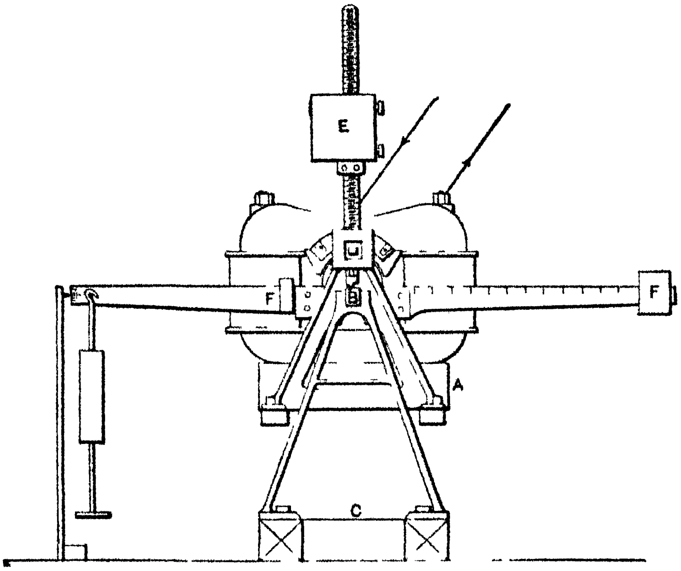

FIG. 234.—CRADLE DYNAMOMETER.

It will be noted that in either case it is necessary to know the efficiency curve of the electrical machine used. In the case of a motor this may be done either by braking the motor or by making electrical measurements and computing the losses. For generators the electrical losses may be found in a similar way, but whenever possible it is simpler to obtain the curves from the builders of the machine, whether motor or generator, and the results will generally be more accurate. This statement applies primarily to machines of medium or large capacity.

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[*] See F. Bedell, Direct and Alternating Current Testing.

The load on a generator may be produced by metallic resistances, batteries of lamps, or water rheostats. Where running water is available, metallic resistances are perhaps the simplest means and may be made of large capacity. In general, however, unless the power output is small, when lamp batteries form a convenient resistance, the water rheostat is perhaps the best for controlling and destroying the electrical output. For design constants for either metallic resistances or water rheostats the reader is referred to electrical handbooks.

167. Methods of Connecting up, etc. — It will not be possible to take up under this head all the types of motors or generators which may be used in practice. Most of the cases will, however, be covered by considering the direct-current shunt motor and generator and the three-phase alternating-current motor and generator.

1. *Motors used as Transmission Dynamometers.* — (*a*) Direct-current Shunt Motor:

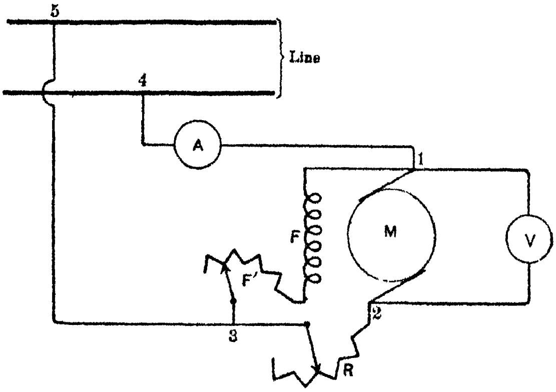

Fig. 235. — D. C. Shunt Motor.

Connection to the lines is made at 4 and 5, Fig. 235. *R* is a starting resistance which should be inserted beyond the point 3 to make certain that the field is always under full excitation. *R* may or may not be used to regulate the speed. It is better for this purpose to insert a variable resistance *F'* in the field circuit. The greater the resistance F', *i.e.*, the weaker the field, the greater the speed. The current should be measured at A, in order to take account of the field current as well. The voltage, however, should be measured between 1 and 2, because the pressure difference at the motor and not in the line is what is wanted. Of course, if R is not used to regulate speed, this precaution is not necessary. The horse-power input to the consumer is $\frac{\text{amperes} \times \text{volts}}{746} \times$ motor efficiency.

(*b*) Alternating-current Motor (Induction Motor):

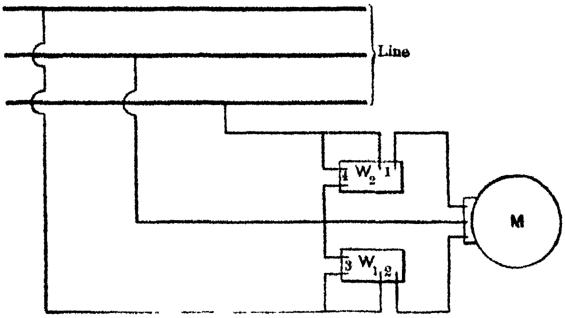

FIG. 236.—A. C. MOTOR.

Connections are shown in Fig. 236 with two wattmeters, W_1 and W_2. Points 1 and 2 indicate the current coil connections to the wattmeters and points 3 and 4 the connections to the potential coils. The power input to the motor is the sum of the wattmeter readings, which multiplied by motor efficiency gives input to power consumer.

2. *Generators used as Brakes.* — (*a*) Direct-current Shunt Generator:

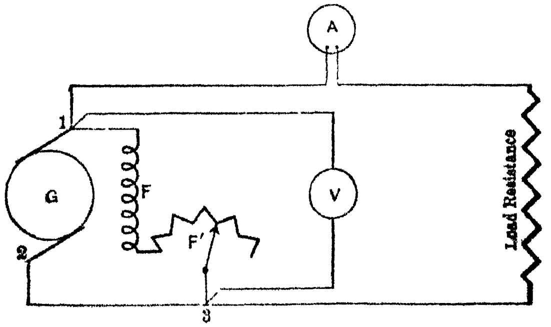

FIG. 237.—D. C. SHUNT GENERATOR.

The connections are very similar to those in Fig. 235. The starting resistance is not used and it is immaterial whether the voltage is measured between points 1 and 2 or 1 and 3. The horse-power input to the generator $= \dfrac{\text{amperes} \times \text{volts}}{746} \div$ generator efficiency.

(*b*) Three-phase Generator:

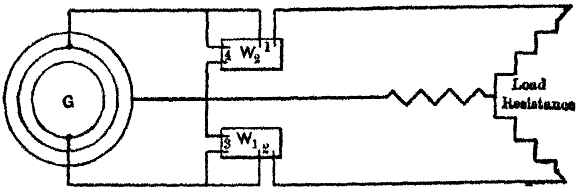

FIG. 238.—A. C. GENERATOR.

Wattmeters are cut in as in Fig. 238. The sum of the two readings is the power generated, which divided by the generator efficiency gives the developed power of the prime mover.

168. The Testing of Belts. — The testing of a belt drive may be for the purpose of determining the efficiency of transmission, the coefficient of friction, the amount of creep or slip, or the working tensions; but it is possible to make a single test with proper arrangement of apparatus which will give sufficient data for the determination and computation of all these items. The belt-testing machine used at Sibley College, described in the following article, is constructed and arranged with this end in view.

For the theoretical consideration of the friction of cords and belting, see Art. 116, p. 238.

169. The Sibley College Belt-testing Machine. — The belt-testing machine illustrated in Fig. 239 is used in the mechanical laboratory of Sibley College. It was designed by Wilfred Lewis of Philadelphia, and used in the tests described in Vol. VII of Transactions of American Society of Mechanical Engineers.*

The belt to be tested is placed on the pulleys *E*, *F*; power is transmitted through the pulley *P* to the Lewis transmission dynamometer and through this to the driving pulley *E*. The driving

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* The student is referred to papers in Transactions of American Society of Mechanical Engineers, Vol. VII, by Wilfred Lewis and Prof. G. Lanza; also to papers in Vol. XII, by Prof. G. Alden; and to the Holman tests in the Journal of the Franklin Institute, 1885.

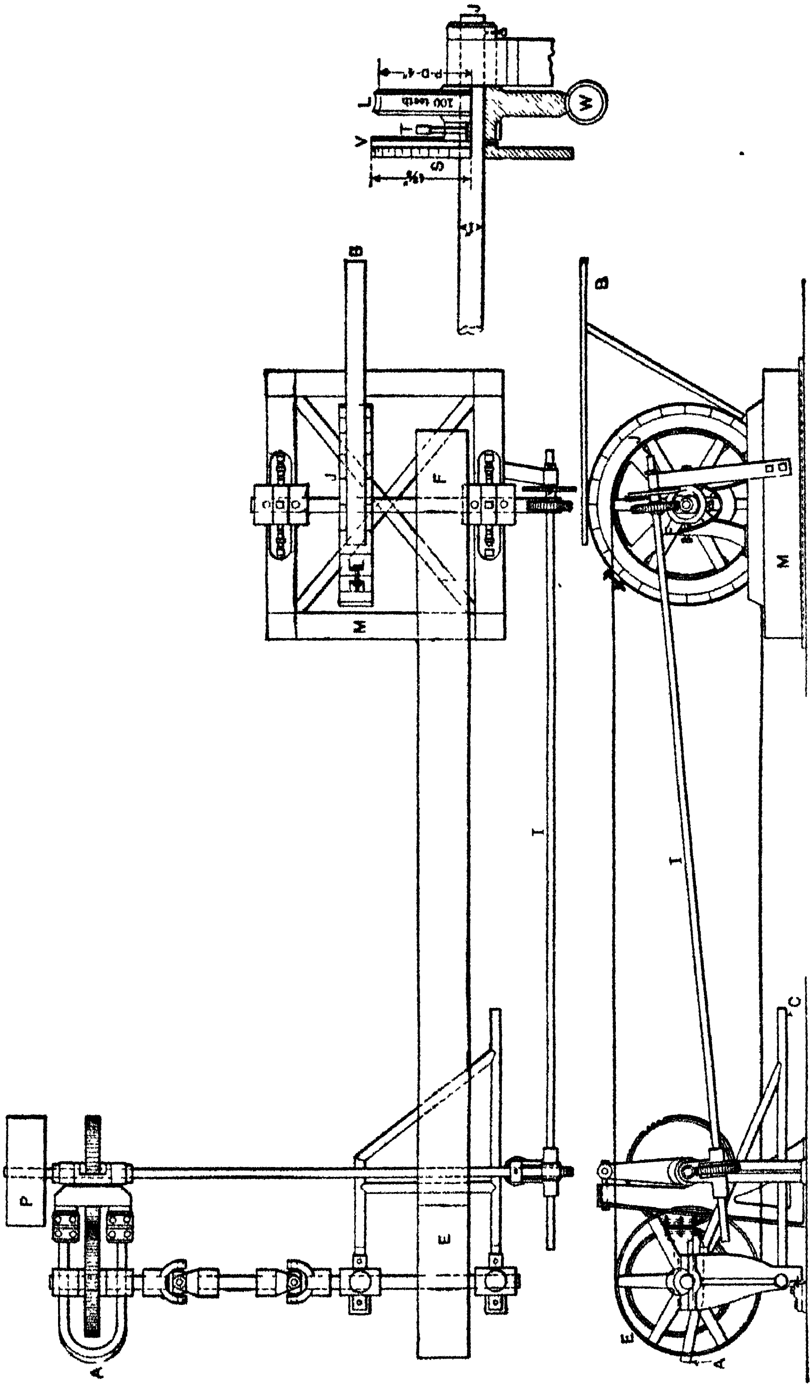

FIG. 239. — SIBLEY COLLEGE BELT-TESTING MACHINE.

shaft is fitted with a universal joint to eliminate transverse strain on the dynamometer, and to allow some freedom of motion transverse to the shaft to the carriage which supports pulley *E*. The tension carriage, so called, pivots near the floor upon sheet-steel fulcrums, like those used in the Emery testing machine. From the driving pulley *E* the power is transmitted to the driven pulley *F*, whose shaft is mounted on solid pedestal bearings on the movable carriage *M*. By shifting the latter along the floor and securing it rigidly, any desired initial tension may be put into the belt. The shaft of *F* also carries a brake wheel *J*, the power transmitted being regulated by means of the Prony brake. The rest of the apparatus consists of a slip disk so constructed that the slip may be read directly in per cent. The auxiliary shaft *I* is driven from the shaft on which pulley *P* is placed through worm gearing having a ratio of 100 to 1. At the opposite end, near the brake carriage (see enlarged detail figure), *I* carries a disk *S*, whose circumference is divided into 100 parts. Next to *S*, but loose on the shaft *I* and driven by a worm *W* on the driven shaft, is the gear *L*. The ratio between *W* and *L* is again 100 to 1. A pointer *V* moving over the circumferential scale of *S* is carried by the hub of *L*, and may be secured in any position desired by the set screw *T*. Now suppose that the machine is running and that *V* is set to any point on *S*. If there is no difference in the position of *V* on *S* when *S* has made one complete revolution, *L* is moving just as fast as *S*, and there is no slip. If, however, *V* has lagged behind say *X* divisions on the scale, the slip will be *X* per cent, because as constructed each division on the scale is equal to a lag of 1 revolution in 100, that is, to a slip of 1 per cent.

Platform scales are provided at *B*, to obtain the net brake load, at *C* to determine the reaction of the tension carriage, and at *A* to find the net load on the dynamometer. In what follows, *B* will be called the brake scale, *C* the tension scale, and *A* the dynamometer scale. Means should also be provided for accurately determining the speed of the driving pulley *E*.

170. Methods of making Tests and Computations. — 1. With test belt off, run the machine at speed desired and determine the zero readings of dynamometer and tension scales. Determine also the zero reading of the brake scale as outlined in Art. 144, p. 276.

2. Put test belt in place under very moderate tension and run for a few turns until it adjusts itself properly to the pulleys. Then shut down and move brake carriage outward until the desired initial tension in the belt is obtained. This is determined by means of the tension scale as follows:

Initial Tension. — In Fig. 240, E is the driving pulley. The belt tensions are T_1 and T_2 respectively, each acting at an arm r. The shaft couple is represented by Px. The vertical arm of the brake carriage is d, the horizontal arm is c and C the net reaction on the tension scale. Then, taking moments about O,

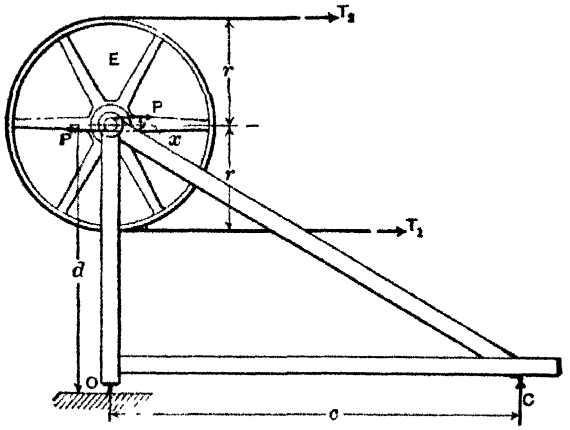

FIG. 240. — TENSION CARRIAGE, BELT-TESTING MACHINE.

$$Cc = T_2 (d + r) + T_1(d - r) + P\left(d + \frac{x}{2}\right) - P\left(d - \frac{x}{2}\right)$$

or

$$Cc = d (T_1 + T_2) - r (T_1 - T_2) + Px.$$

But $r (T_1 - T_2)$ is the couple due to the belt, while Px is the same couple on the shaft. Hence $-r (T_1 - T_2) + Px = 0$, and we have

$$Cc = d (T_1 + T_2)$$

from which

$$T_1 + T_2 = C\frac{c}{d}. \tag{26}$$

But in this case, since the belt is standing still, $T_1 = T_2$ and therefore

$$T = C\frac{c}{2d}.$$

This is the *total* initial tension, T in *each* side of the belt, and is usually divided either by the width of the belt to reduce it to *initial tension per inch of width*, or by the cross-sectional area of the belt to reduce it to *initial tension per square inch of belt.*

3. Start machine and take a series of observations, varying the brake load, *i.e.*, the horse power delivered, from an amount as low as the brake will carry, until the slip becomes excessive, which point is indicated either by screeching and flopping of belt or by belt leaving the driven pulley. For each brake load, as soon as conditions become constant, observe reading of dynamometer scale, tension scale, brake scale, speed of driving shaft, the arc of contact, and slip. To cover the entire field a series of initial tensions should be taken and the same operation repeated for each.

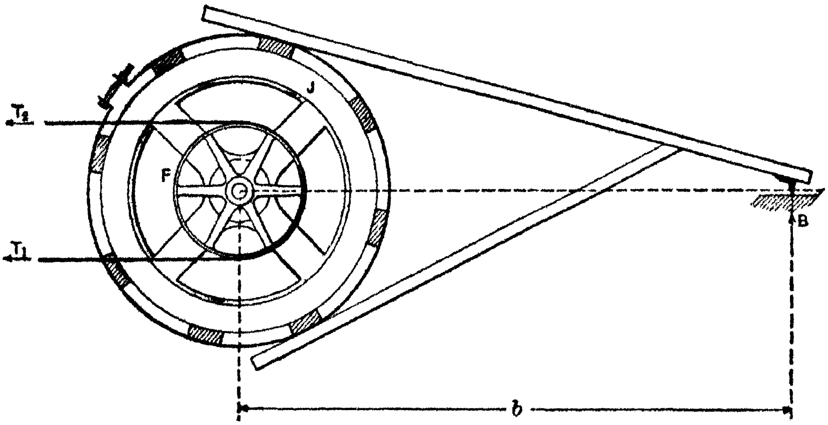

FIG. 241.—BRAKE CARRIAGE, BELT-TESTING MACHINE.

4. *Computations.*—(*a*) *Horse-power Input.*—See description of Lewis dynamometer, Art. 158.

(*b*) *Initial Tension* per inch of width or per square inch of cross section; see 2 above.

(*c*) *Running Tensions,* T_1 in driving or tight side of belt and T_2 in slack side. It was shown under 2 that the sum of the tensions is

$$T_1 + T_2 = C\frac{c}{d}$$

Fig. 241 shows the moments involved in the brake scale mechanism. Here F is the driven pulley, J the brake wheel, the radius of F is $\frac{D_1}{2}$ ft., the length of the brake arm $= b$ ft., and the net brake load $= B$ lbs. With moments above the center of wheel F we have

$$Bb + T_2\frac{D_1}{2} = T_1\frac{D_1}{2},$$

from which

$$T_1 - T_2 = B\frac{b}{\frac{D_1}{2}} = B\frac{2b}{D_1} \tag{27}$$

Combining this with equation (26) above, we finally have

$$T_1 = \frac{C\frac{c}{d} + B\frac{2b}{D_1}}{2}, \tag{28}$$

$$T_2 = \frac{C\frac{c}{d} - B\frac{2b}{D_1}}{2}. \tag{29}$$

These equations give the *total* tensions T_1 and T_2, which are usually reduced to pounds per inch of width or per square inch of cross section by proper division.

(*d*) *Coefficient of Friction, f.*—This may be obtained from equation (7), Art. 116. Usually, however, the equation is simplified to the following form:

$$f = \log_e \frac{T_1}{T_2} \div \theta, \tag{30}$$

where θ is the arc of contact in radians.

(*e*) *Delivered Horse Power* is computed from the constants of the Prony brake and the brake load; see Art. 144.

(*f*) *Efficiency of Transmission*

$$= \frac{\text{Delivered Horse Power}}{\text{Horse-power Input}}.$$

It should be noted here that the efficiency as above computed takes into account the friction of the bearings and is consequently not the net efficiency of transmission by the belt alone.

5. The following form used at Sibley College shows the manner of recording data and of arranging the results.

DEPARTMENT OF EXPERIMENTAL ENGINEERING — SIBLEY COLLEGE.

TEST OF BELTING.

Kind of Belt.................................. Length.. .....ft. Width........in. Thickness........in.
Condition...
Observers...Date..............191

CONSTANTS OF MACHINE.

Diam of Dynamometer Pulley (a)		Diam Driving Pulley (D)		Face of Driving Pulley — inches		Zero of Brake Scale — lbs.	.
Arm of Prony Brake (b)		Diam Driven Pulley (D₁)		Face of Driven Pulley — inches		Zero of Tension Scale, Running	.
Hor Arm on Tension Scale (c)		Circumference Driving Pulley, ft		Kind of Pulley used		Zero of Tension Scale, Standing	.
Vert Arm on Tension Scale (d)		Circumference Driven Pulley, ft		Zero of Dynamometer Scale — lbs		$f = [H \div \text{Log}(T_1 - T_2)] - \theta$	

Run No	R P M of Driving Pulley	R P M of Driven Pulley	Slip per cent	Run Speed of Driver —ft per min	Arc of Contact — Radians.	Arc of Contact — Degrees	Initial Belt Tension Reading	Initial Tension per in of width Net	* Scale Readings in lbs			$Bb + \frac{1}{2}D_1$	$Cc + d$	Belt Tension — Tight Side	Belt Tension — Slack Side	Ratio of Tensions	Coefficient of Friction	Dynamometer Horse Power	Brake H P	Eff = $H + H_1$	Brake H P per inch of width	Max Tension on Tight Side per in of width	Max Working Tension per in. width $(T_1 - T) + \text{width}$		
									Dynamometer	Brake.	Tension														
	n	n_1	s	v	θ	θ_1			A	B	C	$T_1 - T_2$	$T_1 + T_2$	T_1	T_2	$T_1 - T_2$	f	H	H_1						

* Whether net or gross

CHAPTER XI.

HEAT AND THE PROPERTIES OF GASES AND VAPORS.

171. The purpose of this chapter is to bring together in one place all the necessary definitions, conceptions, and laws concerning gases and vapors and thus to avoid loading down the chapters dealing with the testing of prime movers with elementary and theoretical discussions.*

172. **Notation and General Definitions.** — Throughout this Chapter the following notation will be used:

Q = Quantity of heat in British Thermal Units, B.t.u.
E = Work in foot-pounds.
J = Joule equivalent = 778.
$A = \frac{1}{J} = \frac{1}{778} = .1285.$
C_p = Specific heat at constant pressure in B.t.u.
C_v = Specific heat at constant volume in B.t.u.
$\gamma = \frac{C_p}{C_v}.$
p = Absolute pressure in pounds per sq. in.
P = Absolute pressure in pounds per sq. ft.
v = Volume of a gas or vapor in cu. ft.
v' = Volume of one pound of gas or vapor in cu. ft. (= specific volume).
t = Temperature in degrees Fahrenheit.
T = Absolute temperature in degrees Fahrenheit.
W = Weight of gas or vapor involved in a given change or process.

173.. **The Heat Unit and the Mechanical Equivalent of Heat.** — Heat has to be measured by an arbitrary unit, and the unit adopted in the English system is the quantity of heat required to raise one pound of water through one degree Fahrenheit. This is known as the British thermal unit (B.t.u.).

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* The main exception to this statement is the discussion concerning specific heat, in Chap. XXI, in connection with gas-engine testing.

In the metric system the unit is taken as that quantity of heat required to raise one kilogram of water through one degree Centigrade.* This unit is known as the *large calorie,* and is the usual engineering unit, the physicist's unit being the *small calorie,* which is the one-thousandth part of the large calorie. The ratio between the B.t.u. and the large calorie is 1 B.t.u. = .252 kilogram calorie.

It is of the greatest importance in some engineering calculations to be able to transpose heat units into equivalent mechanical units or *vice versa.* The determination of the relation existing between them is entirely experimental, and from the first investigator who determined a definite value for the British thermal unit the conversion factor has obtained the name Joule's equivalent. Joule found the *mechanical equivalent* of one heat unit equal to 772 foot-pounds. This value was later changed to 778 foot-pounds, which is the figure now generally used, although some uncertainty still exists as to the true value. The mechanical equivalent of heat in the metric system is 1 calorie = 424 meter-kilograms.

174. Specific Heat. — Specific heat is generally defined as the quantity of heat required to raise unit weight of a substance through one degree of temperature. In this country the units are one pound and one degree Fahrenheit. From the nature of the definition of the heat unit given above, the specific heat of water is 1.00. The expansion of liquids and solids under a temperature rise of one degree is extremely small, so that practically all the heat furnished goes to increase the intrinsic energy of the substance. There is in the case of such material, therefore, no practical distinction between the specific heat at constant volume, C_v, and that at constant pressure, C_p. With gas the case is different. Here the volume of the gas may be kept constant while it is being heated through one degree or the gas may be allowed to expand in the process. In the first case no external work is done and C_v is consequently always smaller than C_p by the amount of heat required to do the work of expansion against the external pressure.*

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* On account of the fact that the specific heat of water varies slightly with temperature, it becomes necessary in the definition of the heat unit to fix the one degree of temperature through which the heat is supplied. In the older definitions this was taken at from 60 to 61° F. or from 15 to 16° C, but lately the unit is based upon a mean value obtained for the entire range from freezing to boiling points of water.

LAWS OF GASES.

175. Boyle's Law, Charles's Law, etc. — An ideal gas is defined as one which, at constant temperature, follows Boyle's law,

$$pv = \text{constant}. \qquad (1)$$

No gas as far as known follows this law strictly, departing from it especially near the region of liquefaction, but for all engineering purposes and in the range usually covered it expresses pressure and volume relation with sufficient accuracy. Pressure and volume, however, are only two of the criteria which define the state of a gas, the third being temperature. The relation of pressure or volume to temperature is expressed by Charles's law,

$$\frac{p}{T} = \frac{p_1}{T_1} \text{ when the volume remains constant} \qquad (2)$$

or

$$\frac{v}{T} = \frac{v_1}{T_1} \text{ when the pressure remains constant.} \qquad (3)$$

The interrelation of the three variables, p, v, and T, may be expressed by a combination of the laws of Boyle and of Charles as follows:

$$\frac{pv}{T} = \frac{p_1v_1}{T_1} = \text{constant}. \qquad (4)$$

Equation (4) is generally known as the *equation of state* or *condition* of the *ideal gas*.

So far nothing has been said concerning units, and as a matter of fact any units may be used, provided only that p and T are absolute pressure and temperature respectively. The constant in equation (4) will of course have a different value depending upon the units in which p, v, and T are expressed and upon the weight and volume of gas concerned. It has, however, become usual in English practice to express the pressure in pounds per square foot, in which case it is designated by P, to take absolute temperature in degrees Fahrenheit, and to take the volume v equal to that of one pound of the gas under the existing conditions of pressure and temperature, that is = v'. With these units the constant in equation (4) is generally designated by the symbol R and equation (4) becomes

---

* For distinction between *mean* and *instantaneous* specific heat see discussion in Chap. XXI. Heat calculations for a given temperature range should be made by use of the *mean* specific heat for that temperature range, unless the simple assumption that specific heat is constant is made.

$$\frac{Pv'}{T} = R \tag{5}$$

For any one gas, R is constant and is the foot-pounds of work done by one pound of gas when it is heated through one degree Fahrenheit. See Table below for values of R for the most common gases.

176. Specific Heat of Gases. — The distinction between specific heat at constant pressure, C_p, and that at constant volume, C_v, was made in Art. 174. For any given gas the specific heat varies with temperature, increasing as the temperature increases. The laws of this increase are not as yet definitely fixed, and for ordinary engineering calculations it is still usual to consider specific heat a constant quantity for a given gas. Enough information is now, however, available to make such an assumption unwarranted except for rough calculation. The subject is discussed at length in Chap. XXI in connection with gas-engine calculations.

The ratio of the specific heats $\frac{C_p}{C_v}$, which decreases with temperature, is designated by the letter γ. It is in all cases computed from the instantaneous values of specific heat.

The following table gives values of C_p, C_v, γ, and R for a series of the more common gases.

Gas.		Weight per cu ft at 29.92″ Hg 32° F	Specific Heat per lb *		Constant R in equation $\frac{Pv'}{T} = R$.
			C_p	C_v	
Hydrogen	H_2	.00562	3.380	2.380	775.6
Carbon Monoxide	CO	.07807	.243	.174	55.23
Methane	CH_4	.04464	.593	.468	97.25
Ethylene	C_2H_4	.07809	.404	.333	55.23
Acetylene	C_2H_2	.07251	.346	.270	59.13
Butylene	C_4H_8	.15590	.404	.333	. . . .
Nitrogen	N_2	.07831	.243	.173	55.23
Oxygen	O_2	.08921	.217	.153	49.79
Air	—	.08072	.2375	1684	53.75
Water vapor	H_2O	.05016	.455	34	85.58
Carbon Dioxide	CO_2	.12268	.201	.155	35.01

* These are instantaneous values for a temperature of about 60°F.

177. Pressure and Volume Changes in Gases. — (*a*) ***Change at Constant Pressure.***—In the rectangular coördinate system, Fig. 242, let OX = axis of volumes v, and OY = axis of absolute pressure P. Let the point 1 define the initial state of v_1 cubic feet of gas at a pressure P_1 and a temperature T_1. Heat is supplied and the gas is allowed to expand at constant pressure to the point 2, where the state is defined by $P_1v_2T_2$. This change is called *isopiestic* or *isobaric*.

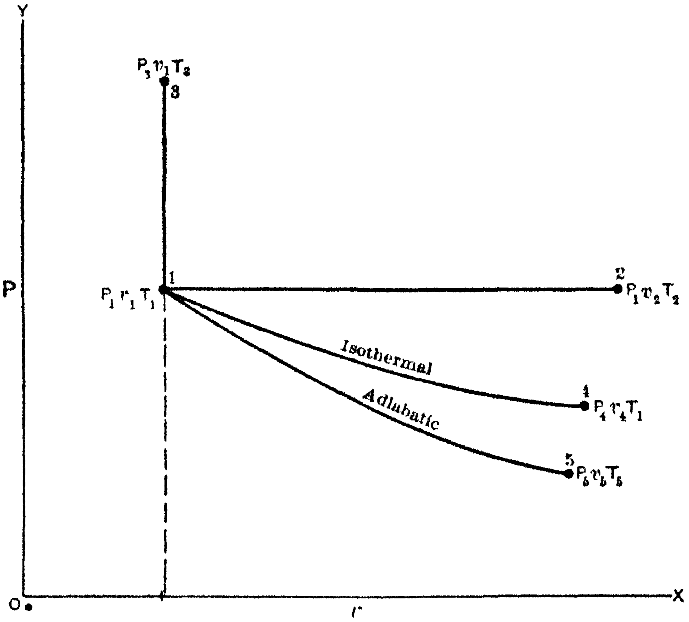

FIG. 242.

Equation of state $\frac{v_1}{T_1} = \frac{v_2}{T_2}$. (6)

Work done $E = P_1(v_2 - v_1)$ ft.-lbs. (7)

Heat supplied $Q = \frac{E}{778} + WC_v(T_2 - T_1)$ B.t.u. (8)

or $Q = WC_p(T_2 - T_1)$ B.t.u. (9)

For the reverse of this change, that is, for a compression from 2 to 1, the same equations hold, except that E and Q will be negative, i.e., work is done upon and heat is rejected by the material changing volume.

(*b*) *Change at Constant Volume.* — With the material again at point 1 in Fig. 242, assume heat supplied and the volume of the gas maintained constant. The pressure rises to the point 3, where the state is defined by $P_3v_1T_3$, according to Charles's law. This change is called *isometric* or *isovolumic.*

Equation of state $\dfrac{P_1}{T_1} = \dfrac{P_2}{T_2}.$ (10)

Work done $E = 0$, since there is no volume change. (11)

Heat supplied $Q = WC_v\ (T_3 - T_1)$ B.t.u. (12)

For the reverse change there is a pressure drop, the quantity of heat, Q, becomes negative, showing that heat must be removed.

(*c*) *Isothermal Change.* — The gas is expanded from the point 1, Fig. 242, to the point 4, heat being supplied to keep the temperature constant. At point 4 the state is defined by $P_4v_4T_1$. This change is called *isothermal,* and the expansion line described is known as an isothermal line. Since T is constant,

Equation of state $P_1v_1 = P_4v_4 =$ constant. (13)

Work done

$$E = \int_{v_1}^{v_4} P\,dv = P_1v_1 \log_e \frac{v_4}{v_1}$$

$$= P_4v_4 \log_e r \text{ ft.-lbs.,} \qquad (14)$$

where r is the ratio of expansion.

Heat supplied $Q = \dfrac{E}{778}$ B.t.u. (15)

For the reverse change, that is, isothermal compression, E and Q become negative, i.e., work must be done upon the gas and heat must be removed.

(*d*) *Adiabatic* (or *Isentropic*) *Change.** — The gas expands from point 1, Fig. 242, to point 5, but no heat is supplied or rejected. This process is known as *adiabatic,* and if reversible, also as *isentropic,* meaning constant *entropy.* For definition of the latter term see page 332. Let the final state of the gas be defined by $P_5v_5T_5$. Since the work done during the change must be done at the expense of the sensible heat in the gas, the temperature must drop, and hence the pressure must drop more rapidly than during an isothermal change.

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* See note, page 344.

Equation of state $$\frac{P_1v_1}{T_1} = \frac{P_5v_5}{T_5}. \tag{16}$$

Equation of adiabatic line $$P_1v_1^{\gamma} = P_5v_5^{\gamma}. \tag{17}$$

From equations (16) and (17) the interrelation of the three variables may be expressed by

$$\frac{T_5}{T_1} = \left(\frac{v_1}{v_5}\right)^{\gamma-1} = \left(\frac{P_5}{P_1}\right)^{\frac{\gamma-1}{\gamma}}. \tag{18}$$

Work done, $$E = \int_{v_1}^{v_5} Pdv = P_1v_1\int_{v_1}^{v_5}\frac{dv}{v^{\gamma}} = \frac{P_1v_1 - P_5v_5}{\gamma - 1}$$

$$= \frac{RT_1 - RT_5}{\gamma - 1}$$

$$= \frac{RT_1\left(1 - \frac{T_5}{T_1}\right)}{\gamma - 1}$$

$$= \frac{RT_1\left[1 - \left(\frac{v_1}{v_5}\right)^{\gamma-1}\right]}{\gamma - 1}$$

$$= \frac{RT_1\left[1 - \left(\frac{P_5}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}{\gamma - 1} \text{ ft.-lbs.} \tag{19}$$

Heat supplied $$Q = 0 \text{ B.t.u.} \tag{20}$$

For the reverse change, i.e., adiabatic compression, E is negative, that is, work is expended in compressing the gas.

(*e*) *Changes represented by Pv^n = constant.*—The equations representing work done, as well as those showing the interrelation of pressure, volume, and temperature for these changes, are the same as those given for the adiabatic change under (*d*) except that n is substituted for γ. As a matter of fact, the adiabatic change is a special case of Pv^n = constant, in which $n = \gamma$. Similarly for the isothermal change $n = 1$, for the change at constant volume $n = \infty$, and for that at constant pressure $n = 0$.

Heat supplied

$$Q = \left[C_v\,(T_1 - T_5) + \frac{E}{778}\right] \text{B.t.u.} \qquad (20a)$$

178. Entropy of Gases. — When heat is supplied to a gas, the result may be a simultaneous change of pressure, volume, and temperature. In other words, the addition of heat may produce, first, additional internal energy of the gas, involving rise of temperature, and second, it may produce an external effect which is equivalent to the work done by the gas expanded between its containing walls. Mathematically expressed, these changes may be written

$$\delta Q = C_v \delta T + \frac{P}{J}\,\delta v'. \qquad (21)$$

By substituting for P in this equation it reduces to

$$\delta Q = C_v \delta T + (C_p - C_v)\,T\frac{\delta v'}{v'}, \qquad (22)$$

and if we write the general equation for entropy,

$$\delta\phi = \frac{\delta Q}{T}, \qquad (23)$$

we will finally have

$$\delta\phi = \frac{\delta Q}{T} = C_v\frac{\delta T}{T} + (C_p - C_v)\,\frac{\delta v'}{v'}, \qquad (24)$$

which is the general equation for entropy of a gas.

According to the simultaneous interrelation between the three factors pressure, volume, and temperature, equation (24) undergoes certain modifications which make it directly applicable to each case.

Case a. Change of Entropy at Constant Volume. — Under this condition

$$(C_p - C_v)\,\frac{\delta v'}{v'} = 0.$$

We shall have from equation (24)

$$\delta\phi = \frac{\delta Q}{T} = C_v\frac{\delta T}{T} \qquad (25)$$

$$\phi_2 - \phi_1 = C_v \log_e \frac{T_2}{T_1}. \qquad (26)$$

Case b. Change of Entropy at Constant Pressure. —From $Pv' = RT$ and $P\delta v' = R\delta T$ we may derive $\frac{\delta v'}{v'} = \frac{\delta T}{T}$, which by substitution in equation (24) gives

$$\delta\phi = \frac{\delta Q}{T} = C_p\frac{\delta T}{T}. \tag{27}$$

$$\phi_2 - \phi_1 = C_p \log_e \frac{T_2}{T_1}. \tag{28}$$

Case c. Change of Entropy with Simultaneous Change of Pressure and Volume. — From equations $Pv'^n = P_1 v'^n_1$ and $P = \frac{RT}{v'}$ we may derive

$$\frac{\delta v'}{v'} = \frac{\delta T}{T(n-1)},$$

which by substitution in general equation (24) changes the latter to the form

$$\delta\phi = \frac{\delta Q}{T} = C_v\frac{\delta T}{T} - (C_p - C_v)\frac{\delta T}{T(n-1)}. \tag{29}$$

This equation may be simplified by the aid of the relation $C_p = \gamma C_v$, the final form being

$$\delta\phi = C_v\frac{\delta T}{T}\left(\frac{n-\gamma}{n-1}\right). \tag{30}$$

$$\phi_2 - \phi_1 = C_v\left(\frac{n-\gamma}{n-1}\right)\log_e \frac{T_2}{T_1}. \tag{31}$$

LAWS OF VAPORS.

179. Vaporization. — The conversion of liquid to vapor takes place at a definite and different temperature for every different pressure, provided the space above the liquid is limited. For the case of *free* evaporation, as for instance in the open cooling tower, the evaporation does not take place at atmospheric pressure, as might be supposed, but at a pressure determined by the temperature of the liquid. This fact must be remembered when making computations for such a case.

To raise unit weight of the liquid to these different temperatures requires in any given case a quantity of heat, called ***heat of the liquid***, equal to

$$q = C_l(T_2 - T_1) \text{ B.t.u.,} \tag{32}$$

in which C_l is the mean specific heat of the liquid between the temperatures T_1 and T_2, T_1 being the temperature of the liquid before the addition of heat. This is generally assumed 32° F. T_2 is the temperature at which vaporization takes place.

It would seem more rational to assume T_1 at the freezing point of the liquid for every substance, although engineering practice still generally assumes T_1 at 32° F., probably because of the much more general use of steam as compared with other vapors. For ammonia vapor, for instance, this basis of computations results in negative heat quantities for the values of q, and in latent heat values r partly negative and partly positive for temperatures below 32°.

After heating the liquid to the temperature of vaporization, the addition of a quantity of heat known as the *latent heat of vaporization* will convert all the liquid to dry saturated vapor at the same pressure and temperature previously possessed by the liquid. This heat is represented by r.

The latent heat of vaporization can be imagined to consist of two parts, the *internal latent heat*, ρ, and the *external latent heat*, Apu.

The internal latent heat is that part of the total latent heat which is used to do the internal work coincident with the molecular rearrangement during the change of state from liquid to vapor.

The external latent heat is that part of the total latent heat which is used for doing the external work necessitated by the enormous volume increase during the change of state.

The *total heat* above 32° per pound of dry saturated vapor is, then,

$$Q_s = \lambda = q + \rho + \text{Apu} = q + r. \tag{33}$$

If the process of vaporization ceases when a fraction, x, of each pound of liquid has been converted into vapor, the mixture is known as *wet saturated vapor*, and the heat above 32° per pound of mixture is

$$Q_w = q + x(\rho + \text{Apu}) = q + xr. \tag{34}$$

As the pressure and the quantities q, ρ, r, Apu, and λ all vary with the temperature, it is customary to tabulate them and other useful properties of the material in so-called *vapor tables*. See the Steam and Ammonia Vapor Tables in the Appendix.

180. Superheating and Specific Heats of Vapors.—After vaporization, as outlined in Art. 179, is complete, the further addition of heat results in raising the temperature above that corresponding to the pressure, if liquid were present. This is known as *superheating*. Vapor may be superheated at constant volume, at constant pressure, or under a simultaneous change of both pressure and volume. The usual method, however, is practically that at constant pressure, and this will therefore be the only case necessary to consider.

Experiment shows that the specific heat of vapors varies with pressure and temperature, and recently fairly accurate determinations of this quantity have been made in the case of water vapor (steam). Among these the results of Knoblauch and Jakob, slightly modified at low pressures and at temperatures near saturation by Marks and Davis,* seem to be the most reliable. Fig. 243 † shows a graphical representation of Knoblauch and Jakob's results so modified. The ordinates of this diagram represent what are called " instantaneous " specific heats, that is, the progressive values which vary from degree to degree at constant pressure and from pressure to pressure at constant temperature. In Fig. 243 the curve AB may be called a saturation curve, since it is a curve obtained by plotting corresponding conditions of saturated steam.

A more useful way of plotting specific heats, however, is to plot the " mean " specific heat for given temperature ranges, because most engineering calculations in this field call for the determination of heat content of superheated steam at a certain definite pressure and temperature.

This may be done by means of the data for superheated steam contained in the Tables of Marks and Davis. The result for a number of pressures is shown in Fig. 244. Thus for a pressure of 100 lbs. absolute and a degree of superheat equal to 200°, the mean specific heat is found to be $C_p = .515$. Hence the B.t.u. represented by this degree of superheat will be $200 \times .515 = 103$ B.t.u.

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* Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam, L. S. Marks and H. N. Davis, Longmans, Green & Co.

† Tables and Diagrams, p. 97.

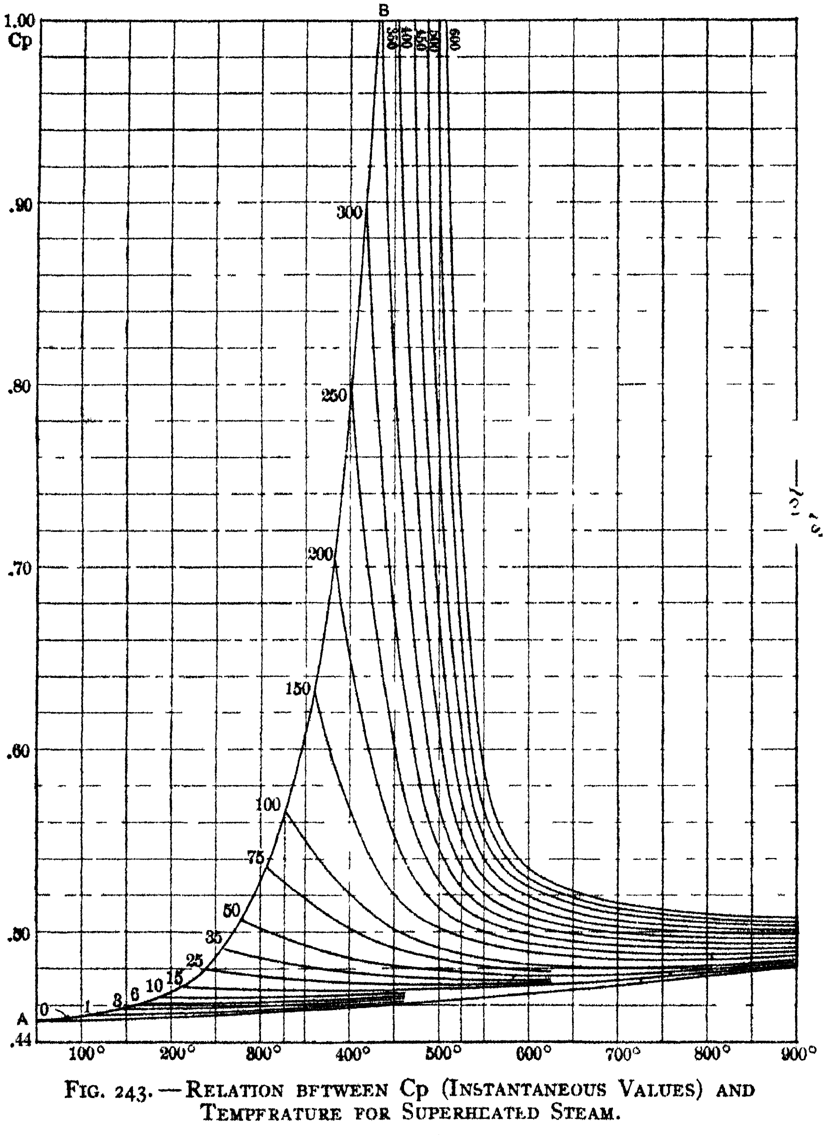
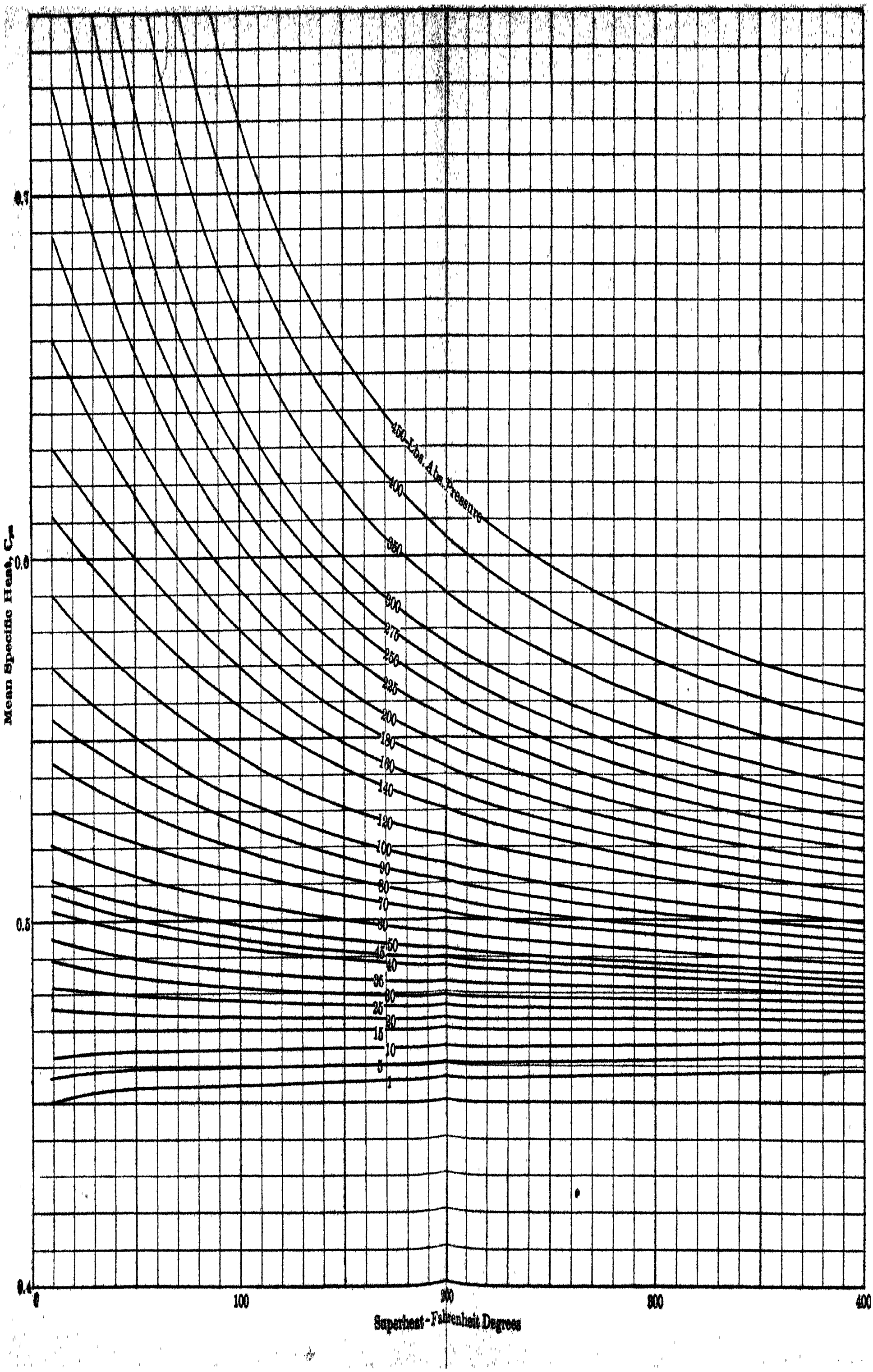


FIG. 243.—RELATION BETWEEN Cp (INSTANTANEOUS VALUES) AND TEMPERATURE FOR SUPERHEATED STEAM.

(The numbers on the curves represent absolute pressure in lbs./sq inch.)

To obtain the same result from the diagram of Fig. 243 we would have had to proceed as follows: The saturation temperature of steam at 100 lbs. absolute is 327.8° F. (from Steam Table, Appendix).


Mean Specific Heat, C_p
0.7
0.6
0.5
0.4
0
100
200
300
400
Superheat - Fahrenheit Degrees
450 Lbs. Abs. Pressure
400
350
300
275
250
225
200
180
160
140
120
100
90
80
70
60
50
45
40
35
30
25
20
15
10
5
1

With 200° of superheat the steam temperature would then be 527.8° F. It will next be necessary to obtain, in Fig. 243, the mean ordinate for that part of the C_p-curve for 100 lbs. between the abscissæ 327.8 and 527.8. This will be found to be .515, but the great advantage of Fig. 244 is at once apparent.

181. Use of Steam Tables and Diagram.* —Table No. I in the Appendix is computed with steam temperatures as a basis and gives the values of the various quantities from 32° to 165° F. It is intended to be used mainly for condenser work, the absolute pressure range covered being from .2″ to 10.86″ Hg.

Table No. II is based on pressures and will serve for all ordinary steam calculations.

The items in both tables are clearly enough set forth in the headings to the columns and need no further explanation.

The diagram, Fig. 245, is of course constructed from steam table data, and the legends and directions it contains should be sufficient to explain its meaning. Some of its uses will be apparent in the following examples:

Example 1. — Steam pressure 125 lbs. absolute, feed water temperature 72°, quality of steam 95 per cent, find total heat per pound of steam above feed water temperature.

General equation:

$$\text{Heat per pound of steam} = [xr + q - (t - 32)] \text{ B.t.u.}$$

From Table II, for 125 lbs. pressure, $r = 874.7$, $q = 315.5$, therefore heat per pound of steam

$$= [(.95 \times 874.7) + 315.5 - (72 - 32)] = 1106.4 \text{ B.t.u.}$$

Example 2. — How much heat is required to heat boiler feed water from 75° to vaporization at a boiler pressure of 150 lbs. by gauge?

For the solution of this problem it is necessary to know the barometric pressure, since the tables are based on absolute pressure. Assume that the barometer stands at 29.3″, equivalent to a pressure of 14.4 lbs. The absolute pressure will then be 150.0 + 14.4 = 164.4 lbs.

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* Both tables and diagrams were taken from the "Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam," by special permission of the authors, Prof. L. S. Marks and Mr. H. N. Davis, and of the publishers, Longmans, Green & Co. The tables were much shortened, and this is true especially of the temperature tables, which in the original covered the range from 32° to 689°. The pressure table is cut down less. The diagram was reduced from a large chart approximately 16 × 20 inches. The book mentioned also contains an extended table of the properties of superheated steam.

From Table II:

At 165 lbs. absolute,	$q = 338.2$ B.t.u. above 32°.
At 164 lbs. absolute,	$q = \underline{337.7}$ " " " ".
Difference,	.5

Hence at 164.4 lbs., $q = 337.7 + (.4 \times .5) = 337.9$ B.t.u. above 32°. From this value must be subtracted the value of q for feed water at 75°. This is best found from Table I, from which $q = 43.05$ B.t.u. Hence the net amount of heat required is in this case equal to $337.9 - 43.05 = 294.85$ B.t.u.

It is quite customary for ordinary calculations, especially if no temperature table is available, to assume the specific heat of water equal to 1.0, in which case the heat in the feed water at 75° above 32° would be 43 B.t.u., and the heat required for the case under discussion would then have been $337.9 - 43 = 294.9$ B.t.u. It will be noted that the difference is very small.

Example 3. — The total heat contained in a pound of steam is found to be 1175 B.t.u. The absolute pressure is 180 lbs. Find the quality of the steam. We have

$$xr + q = 1175.$$

From the steam table, for 180 lbs., $r = 850.8$, $q = 345.6$, hence

$$850.8\,x + 345.6 = 1175,$$

from which

$$x = 97.5 \text{ per cent}$$

The same result could have been obtained very quickly from the diagram, Fig. 245, by following the 1175 B.t.u. line to the right to its intersection with the 180 lbs. pressure line. This point will be found to lie halfway between the 97 and 98 per cent quality lines.

Suppose, next, that at the same absolute pressure the steam had contained 1260 B.t.u. per pound; find x. Here

$$850.8\,x + 345.6 = 1260,$$

and

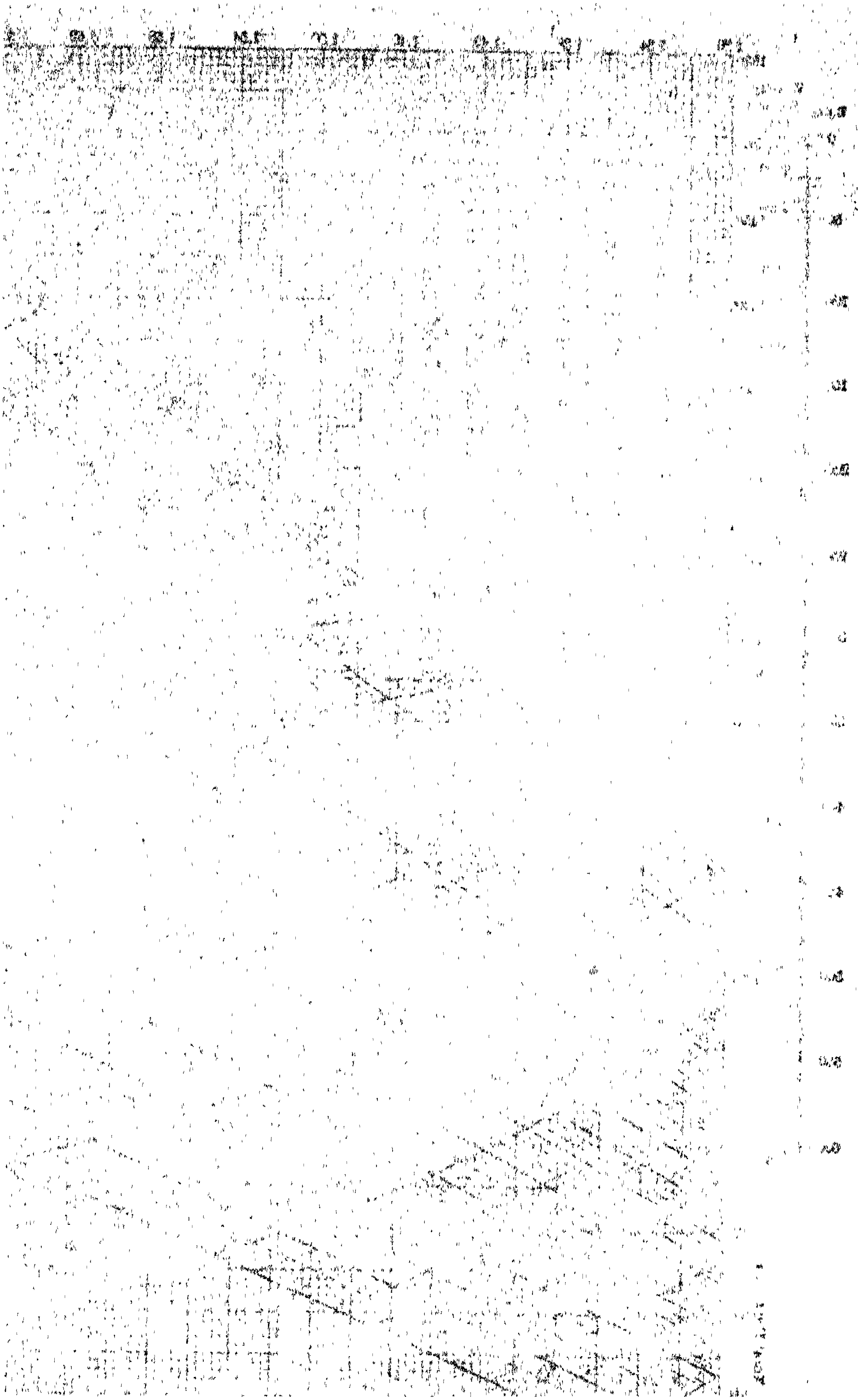
$$x = 1.075.$$

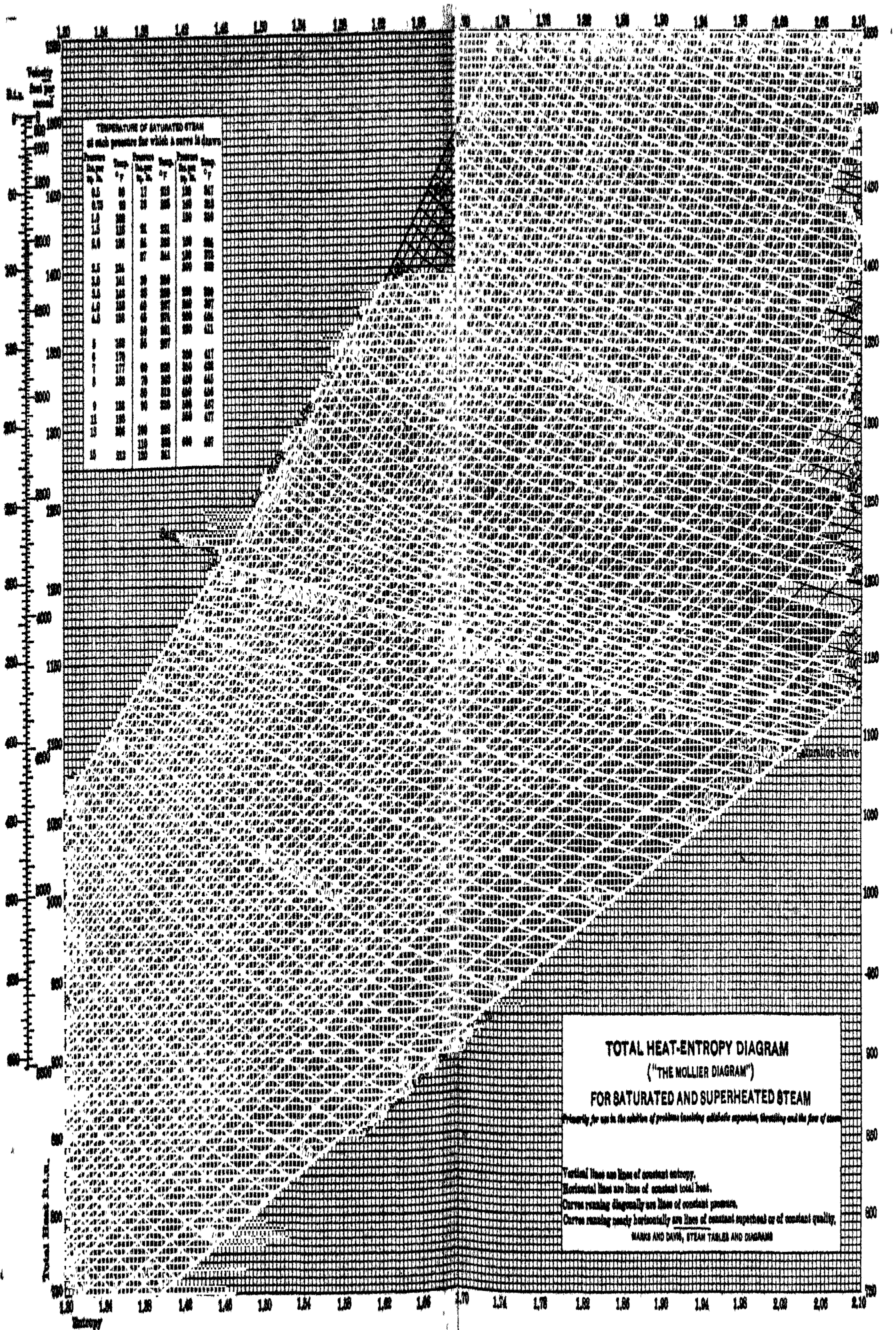
Evidently the steam is superheated, and the amount of superheat should next be expressed either in B.t.u. or in degrees. To obtain the B.t.u. of superheat, the quickest way is to subtract from 1260 B.t.u. the total heat in dry and saturated steam at 180 lbs. absolute, equal to 1196.4 B.t.u. from the table. This shows 63.6 B.t.u. of superheat. The next step is to determine the degree of superheat, i.e., the steam temperature. We have

$$C_p\,(T_2 - T_1) = \text{B.t.u. of superheat},$$

where

C_p = mean specific heat in the range T_1 to T_2,
T_2 = actual temperature of steam,
T_1 = saturation temperature of steam.





TEMPERATURE OF SATURATED STEAM
at each pressure for which a curve is drawn

Pressure lbs. per sq. in.	Temp. °F	Pressure lbs. per sq. in.	Temp. °F	Pressure lbs. per sq. in.	Temp. °F
0.5	80	17	219	130	347
0.75	92	20	228	140	353
1.0	102			150	358
1.5	116	21	231		
2.0	126	24	238	160	364
		27	244	180	373
2.5	134			200	382
3.0	141	30	250		
3.5	148	35	259	220	390
4.0	153	40	267	240	397
4.5	158	45	274	260	404
		50	281	280	411
5	162	55	287		
6	170			300	417
7	177	60	293	350	431
8	183	70	303	400	445
		80	312	450	456
9	188	90	320	500	467
11	198			550	477
13	206	100	328		
		110	335	600	486
15	213	120	341		

FIG. 245.—TOTAL HEAT-ENTROPY (MOLLIER) DIAGRAM FOR STEAM.

In this case,
$$C_p\,(T_2 - 373.1) = 63.6,$$
$$T_2 = \frac{63.6}{C_p} + 373.1.$$

If the only information available concerning the interrelation of C_p and T_2 is that contained in the curves of Fig. 244, the last equation will have to be solved by trial, since both C_p and T_2 are unknown. From Fig. 244 it will be found that when the degree of superheat = 111, $C_p = .573$ for 180 lbs. absolute, and this pair of values closely satisfies the above equation. Hence the steam temperature is 373.1 + 111 = 484.1° F.

Again the diagram, Fig. 245, would have given this result without computation, for following the 1260 B.t.u. line to the right until it crosses the 180 lbs pressure line, it will be seen that the point of intersection is just about halfway between the 100° and the 120° lines of superheat.

Example 4. — The vacuum gauge on a condenser shows 26″ Hg, while the temperature in the condenser is 110°. The barometer shows 29″ at the same time. What proportion of the absolute pressure in the condenser is due to the presence of air?

A temperature of 110° in the condenser, if steam alone were present, calls for an absolute pressure of 2.589″ Hg, according to the temperature table in the Appendix. The absolute pressure really is 29 − 26 = 3″ Hg, and the excess pressure, or .411″ Hg, is therefore due to the presence of air.

Example 5. — One pound of steam at an absolute pressure of 150 lbs. and a temperature of 460° F. is expanded isentropically to a pressure of 30 lbs. absolute. What is the state of the steam at the end of the expansion and what is the difference in heat content of steam at beginning and end?

The saturation temperature of the steam is 358.5° F., hence the steam is superheated to an extent of 460 − 358.5 = 101.5°. The problem is best solved by means of the diagram, Fig. 245. Since in this diagram the abscissæ represent entropy, an isentropic change is represented by a vertical line. Starting, therefore, on the 150 lbs. pressure line at a point representing 101.5° of superheat, drop a vertical line to an intersection with the 30 lbs. pressure line. The point of intersection indicates a quality of 95.1 per cent, the steam having changed its state from that of superheat to 4.9 per cent wet. For 150 lbs. absolute pressure and 101.5° of superheat, the B.t.u. scale at the left shows a heat content of 1255 B.t.u. per pound. At 30 lbs. and 95.1 per cent quality, the scale shows a heat content of 1118 B.t.u. Hence the difference of heat content is in this case 1255 − 1118 = 137 B.t.u.

Example 6. — Find the heat required to superheat 1 pound of steam from 425 to 500° F. at a constant pressure of 200 lbs. absolute.

The saturation temperature for 200 lbs. is 381.9°, hence, the above temperature ranges call for a degree of superheat increasing from 43.1° to 118.1° F. Fig. 244 shows that at 43.1° and 200 lbs. mean $C_p = .642$, hence the heat required to superheat up to that point is 43.1 × .642 = 27.67 B.t.u. At

118.1° and 200 lbs., $C_p = .582$, hence the heat required up to this temperature is $118.1 \times .582 = 68.72$ B.t.u. The net expenditure of heat to raise the temperature from 425° to 500°, therefore, is $68.72 - 27.67 = 41.05$ B.t.u. per pound. The same result would have been obtained much more quickly from the diagram, Fig. 245, by reading off the difference in the heat content of steam at 43° and 118° of superheat, following the constant pressure line 200, although this chart cannot give the result quite so closely.

182. Entropy of Vapors. — Defining entropy by the equation

$$d\phi = \frac{dQ}{T}, \tag{35}$$

a finite entropy change must be

$$\Delta\phi = \int \frac{dQ}{T}. \tag{36}$$

During the heating of a liquid preparatory to vaporization

$$dQ = C_l dT, \tag{37}$$

where C_l = specific heat of the liquid. Substituting this value of dQ in equation (36), we have the entropy change of the liquid

$$\Delta\phi_l = \int_{T_1}^{T_2} C_l \frac{dT}{T}, \tag{38}$$

which, in case C_l is assumed constant or taken as the mean value through the temperature range, can be written

$$\Delta\phi_l = C_l \int_{T_1}^{T_2} \frac{dT}{T} = C_l \log_e \frac{T_2}{T_1}. \tag{39}$$

In the so-called "entropy of the liquid" in the steam tables, as computed above 32°, the value of T_1 in the above equation is equal to 492, while T_2 is the temperature of vaporization.

During vaporization of a liquid at constant pressure the temperature remains constant, and the heat added is the latent heat of vaporization, r, so that the entropy change during vaporization is

$$\Delta\phi_v = \int \frac{dQ}{T} = \frac{r}{T}. \tag{40}$$

If vaporization is incomplete, i.e., if the vapor formed has the quality, x, less than unity, the heat added during vaporization is only xr, and the entropy change is

$$\Delta\phi_{xv} = \int \frac{dQ}{T} = \frac{xr}{T}. \tag{41}$$

The total entropy change, therefore, experienced by a liquid when heated from any temperature T_1, and vaporized at a given constant pressure at a temperature T_2, will be

$$\Delta\phi_x = C_l \log_e \frac{T_2}{T_1} + \frac{xr}{T_2}. \tag{42}$$

If the vapor is dry and saturated, this equation becomes

$$\Delta\phi_{sat} = C_l \log_e \frac{T_2}{T_1} + \frac{r}{T_2}. \tag{43}$$

If the vapor is next superheated at constant pressure (usual case) dQ in equation (36) may be written $= C_p dT$, and

$$\begin{aligned}\Delta\phi_d &= \int_{T_2}^{T_3} C_p \frac{dT}{T} \\ &= C_d \log_e \frac{T_3}{T_2}\end{aligned} \tag{44}$$

where C_d is the mean specific heat for the temperature range $T_3 - T_2 = T_d$. The total entropy change per pound of liquid changed from a temperature T_1 into superheated vapor at a temperature T_3 will be

$$\Delta\phi_s = C_l \log_e \frac{T_2}{T_1} + \frac{r}{T_2} + C_d \log_e \frac{T_3}{T_2} \tag{45}$$

provided the vapor has been superheated at constant pressure.

The vapor table for steam in the Appendix gives the entropy of the liquid, the entropy of vaporization, and the total entropy of dry and saturated steam. If the steam is superheated, the third member in equation (45) will have to be computed.

183. The Temperature-Entropy Diagram for Steam. — In a rectangular system of coördinates, Fig. 246, let the abscissæ represent entropy, designated by ϕ, and the ordinates represent absolute temperatures, T. Entropy changes of the liquid will in such a diagram be represented by a line a-b-b_1, etc., called the *liquid* or *water curve*, the data for which line may be obtained either by solving equation (39) or by use of the steam table in the Appendix. The diagram is constructed for one pound of water. Suppose that vaporization begins at the point b, where the absolute temperature is 653.2°, corresponding to an absolute pressure of 10 lbs. At this point the entropy of the liquid above 32° is 0.2832. See steam table. During vaporization the temperature is constant, until the point *c* is reached, where all of the water has been evaporated. The heat of vaporization during this change is 982 B.t.u., and the entropy change, therefore, is $\frac{982}{653.2} = 1.5042$, which, added to the entropy at *b*, makes a total entropy change above 32°, equal to 1.7874. If only a part of the water is vaporized, only a proportionate part

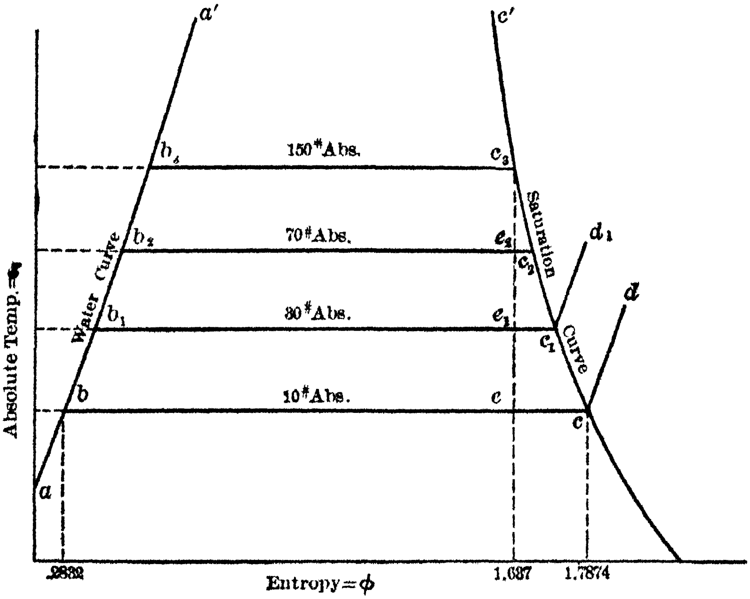


FIG. 246.—ENTROPY DIAGRAM FOR STEAM.

of the heat of vaporization is used. For instance, assume that the quality of the vapor is 90 per cent. Then the heat rendered latent in the vaporization process = .90 × 982 = 883.8, and the entropy change, therefore, is $\frac{883.8}{653.2} = 1.3538$. The total entropy change above 32°, therefore, is 1.6370. This locates the point *e* on the diagram. The line *be* is equal to .9 of the line *bc*, and, in fact, the quality of steam in this diagram may be determined by finding the ratio of actual length of the vaporization line to what would be the length if vaporization were complete.

If, on reaching the point c, the vapor is next superheated, the temperature again rises with an increase in ϕ as represented by the line *cd*, the data for which may be computed from equation (44).

For any other pressure, as 30 lbs., the process will be determined by the line $ab_1c_1d_1$, the data for which is obtained in exactly the same manner.

If the points c, c_1, c_2, . . . are joined by a smooth curve, the resulting line is the so-called ***saturation curve*** for steam.

In the region to the right of this line the material must be superheated, while to the left it must consist of wet vapor until the line abb_1 is reached, where all the vapor is condensed.

184. Pressure and Volume Changes in Vapors. — (*a*) *Change at Constant Pressure.*

Case of Saturated Vapor. — During a volume change at constant pressure, with a saturated vapor, the temperature remains constant. Hence, for such a vapor the isothermal line and the isobar are identical. Also, since a change of volume at constant pressure with a saturated vapor can only occur during vaporization or condensation, every such change is accompanied by a change in the quality of the vapor.

Equation of the line, $P =$ constant. (46)

Work done, $E = \mathrm{P}\ (v_2 - v_1)$ ft. lbs. (47)

Heat supplied during an expansion from v_1 to v_2 in which the quality of the vapor changes from x_1 to x_2

$$= Q = (q_2 + x_2r_2) - (q_1 + x_1r_1) \text{ B.t.u.} \quad (48)$$

in this case $q_2 = q_1$, and $r_2 = r_1$, hence,

$$Q = r(x_2 - x_1) \text{ B.t.u.} \quad (48a)$$

Case of Superheated Vapor. — The conditions accompanying a change of volume at constant pressure in a superheated vapor are best defined by the lines representing the entropy changes during superheating in Fig. 246. Accordingly, any such change is accompanied by a rise or drop in temperature, depending upon whether there is expansion or contraction, and such temperature changes are perhaps most easily considered by the construction of a curve similar to *cd*. In pressure-volume coördinates, the following approximate formula of Tumlirz may be used in the case of steam:

$$v = .5962 \frac{T}{p} - .256, \tag{49}$$

in which v = specific volume of the superheated steam,

T = absolute temperature in F degrees,

p = absolute pressure in lbs./sq. in.

Equation for the line, P = constant. (50)

Work done, $E = P(v_2 - v_1)$ ft.-lbs. (51)

Heat supplied during an expansion from v_1 to v_2,

$$Q = C_p(T_2 - T_1) \text{ B.t.u.}, \tag{52}$$

where T_1 and T_2 may be obtained by use of Tumlirz's equation.

(*b*) *Constant Volume Changes.*

Case of Saturated Vapor:

Equation of line v = constant. (53)

Work done, E = 0 ft.-lbs. (54)

Heat supplied $Q = (q_2 + x_2\rho_2) - (q_1 + x_1\rho_1)$ B.t.u. (55)

Case of Superheated Vapor:

Equation of line, v = constant. (56)

Work done, E = 0 ft.-lbs. (57)

Heat supplied, $Q = C_v(T_2 - T_1)$ B.t.u. (58)

Constant volume changes actually occurring are, however, usually cases of *changing mass*, and the equations above do not then apply. (Example, toe of indicator card after release in steam or gas engines.)

(*c*) *Isentropic Change.** —An idea of the volume, pressure, temperature, and quality changes which accompany an isentropic change, say expansion, in a vapor, is best gained from a preliminary study of the entropy-temperature diagram. In Fig. 246, aa' is the liquid line for steam and cc' the saturation line for steam. Since, during a reversible adiabatic change, there is no change in the total entropy of the vapor, such a change will be indicated by a straight line parallel to the axis of temperatures. For instance, starting at the point c_3, on the 150 pound pressure line, an isentropic expansion to 10 lbs. pressure is indicated by the line c_3e. During this change the quality of steam, which was equal to unity at c_3, steadily decreases until at 10 lbs. pressure it is measured by the ratio $\frac{be}{bc}$. Similarly, at any other pressure, as 70 lbs., the quality will be $\frac{b_2e_2}{b_2c_2}$, and it will be noticed that the quality decreases as the pressure drops.

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* An *adiabatic* process is one in which interchange of heat, *as heat*, between the working medium and other bodies does not occur, —in other words the working body is thermally isolated. An *isentropic* change is defined as one during which the entropy remains constant. It has been held that, in the case of vapors, these two terms are not necessarily synonymous. It is clear however that in the great majority of cases, when the engineer uses the word adiabatic, he refers to the condition of constant entropy, *i.e.* to the isentropic process.

The quality variations during isentropic expansions vary with different initial qualities, and no simple law which expresses these quality changes can be given. For this reason it is again best to resort to the $T\phi$ diagram when studying any particular case.

This diagram shows the entropy, temperature, pressure, and quality changes, but does not directly show the volume changes. Since most of the actual engineering calculations are based on pressure-volume changes, it becomes necessary to find a means of determining the volumes corresponding to the different points e, e_1, e_2, etc.

Neglecting the volume occupied by water, which in all real cases is almost vanishingly small, the volume occupied by a pound of wet steam is to the volume occupied by a unit weight of dry and saturated steam at the same pressure as the quality of the wet steam is to unity, that is,

$$\frac{v_x}{v} = \frac{x}{1},$$

where v_x is the volume occupied by one pound of steam at quality x, and v is the volume occupied by one pound of dry and saturated steam at the same pressure, that is, the specific volume, which latter may be found from steam tables.

From this it follows that the volume occupied by steam with conditions shown by any point such as e in the $T\phi$ diagram, Fig. 246, is equal to the corresponding quality multiplied by the specific volume, that is,

$$v_e = \frac{be}{bc}v = x \text{ times specific volume.}$$

It is therefore possible to plot a Pv curve representing isentropic changes of saturated steam by means of the $T\phi$ diagram and the specific volumes given in the steam tables.

For superheated conditions the isentropic change must also be represented by a vertical line in the $T\phi$ diagram. Here, again, this diagram shows temperature and pressure variations accompanying the isentropic change, but it gives no indication of the volumes attained. These are best computed from the equation of Tumlirz (49).

Equation of the line:

For isentropic expansion of steam, approximate formulæ of the form Pv^n = constant, have been developed.

For saturated steam with initial quality, x, between 0.7 and 1.0,

$$Pv^{1.035 + 0.1x} = \text{constant} \tag{59}$$

For steam remaining superheated

$$Pv^{1.34} = \text{constant.} \tag{59 a}$$

Work done is

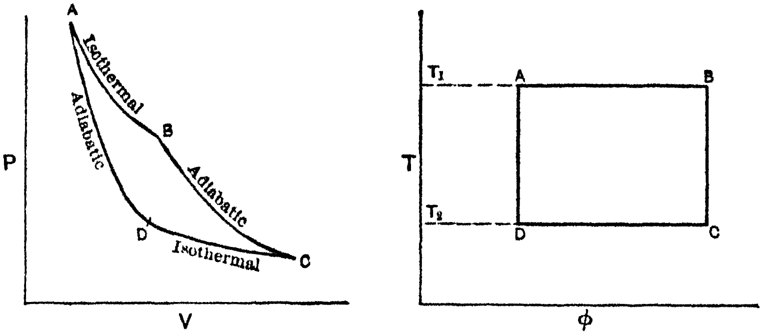
$$E = \frac{P_1v_1 - P_2v_2}{(1.035 + 0.1\,x) - 1} \text{ ft.-lbs.} \begin{Bmatrix}\text{for saturated steam with initial}\\ \text{qualities between 0.7 and 1.0}\end{Bmatrix}. \tag{60}$$

$$E = 778\,(q_1 + x_1r_1 - q_2 - x_2r_2) \text{ ft.-lbs.} \begin{Bmatrix}\text{for saturated steam with}\\ \text{any initial quality}\end{Bmatrix}. \tag{61}$$

In all cases the heat supply is Q = 0 B.t.u. by definition of an adiabatic.

185. Cycles and Their Efficiencies. — The cycles discussed under this head are purely theoretical, so that the efficiency formulæ developed and the efficiences computed from them show the highest theoretically obtainable by ideal engines which operate upon these cycles. In any actual case, owing to various losses, the real cycles only approximate the theoretical cycles, so that the theoretical efficiency may serve as a standard and the ratio of the efficiency actually obtained to the theoretical efficiency may be taken as a measure of the degree of perfection of actual operation. This ratio has in some cases a specific name. Thus, in the case of the reciprocating steam engine, it has been called the "cylinder efficiency;" in the case of the turbine the name "potential efficiency" has been used.

Gas Cycles. — (*a*) *Carnot Cycle,* shown in Figs. 247 and 248.



Figs. 247, 248. — Carnot Cycle for Gas.

$$\text{Heat supplied per lb. of gas} = P_A V_A \log_e \frac{V_B}{V_A} = RT_1 \log_e r \text{ ft.-lbs.} \quad (62)$$

$$= T_1 (\phi_B - \phi_A) \text{ B.t.u.} \quad (63)$$

$$\text{Heat discharged} = P_C V_C \log_e \frac{V_C}{V_D} \text{ ft.-lbs.} = RT_2 \log_e r \text{ B.t.u.} \quad (64)$$

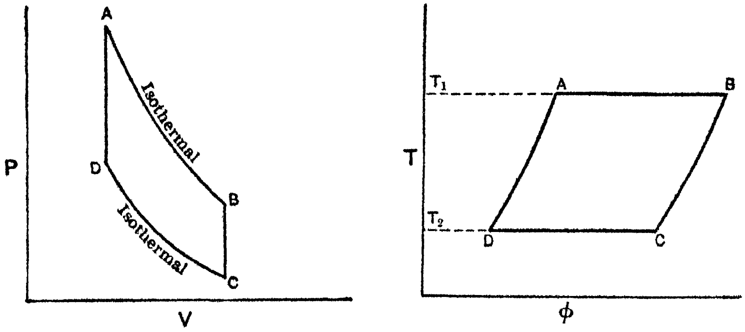
$$= T_2 (\phi_C - \phi_D) \text{ B.t.u.} \quad (65)$$

$$\text{Work done} = R (T_1 - T_2) \log_e r \text{ B.t.u.} \quad (66)$$

$$= (T_1 - T_2) (\phi_B - \phi_A) \text{ B.t.u.} \quad (67)$$

$$\text{Efficiency} = \frac{R (T_1 - T_2) \log_e r}{RT_1 \log_e r} = \frac{(T_1 - T_2) (\phi_B - \phi_A)}{T_1 (\phi_B - \phi_A)} = \frac{T_1 - T_2}{T_1}. \quad (68)$$

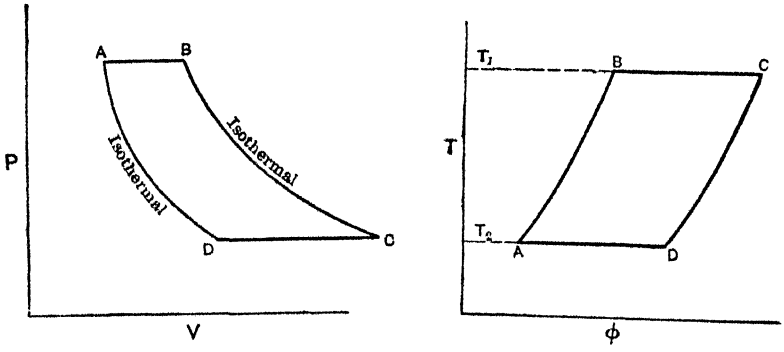
(*b*) *Stirling Cycle* (used in hot-air engines). Figs. 249 and 250.



FIGS. 249, 250.—STIRLING CYCLE.

All the formulæ for this cycle are the same as those developed above for the Carnot cycle. The heat discharged along BC is stored in the regenerator, which is a part of the engine, and is restored to the working substance along DA.

(*c*) *Ericsson Cycle* (used in hot air engines). Figs. 251 and 252.

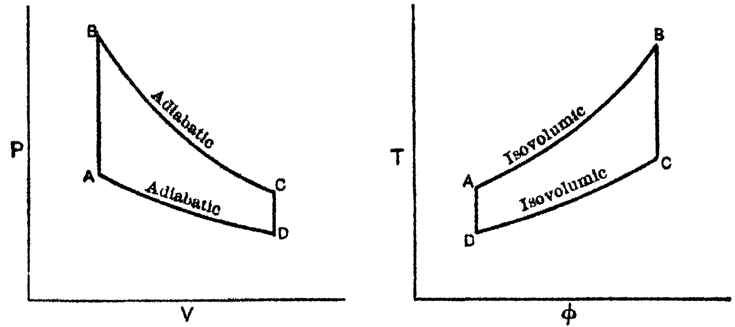


FIGS. 251, 252.—ERICSSON CYCLE.

The formulæ for this cycle also are the same as for (*a*) or (*b*).

The action of the regenerator along AB and CD is the same as under (*b*).

(*d*) *The Otto Cycle.* Figs. 253 and 254.



FIGS. 253, 254.—OTTO CYCLE.

Heat supplied per lb. of gas (along AB)

$$= \int_{T_A}^{T_B} C_v dT = C_v (T_B - T_A) \text{ B.t.u.} \qquad (69)$$

Heat discharged per lb. of gas (along CD)

$$= \int_{T_D}^{T_C} C_v dT = C_v (T_C - T_D) \text{ B.t.u.} \qquad (70)$$

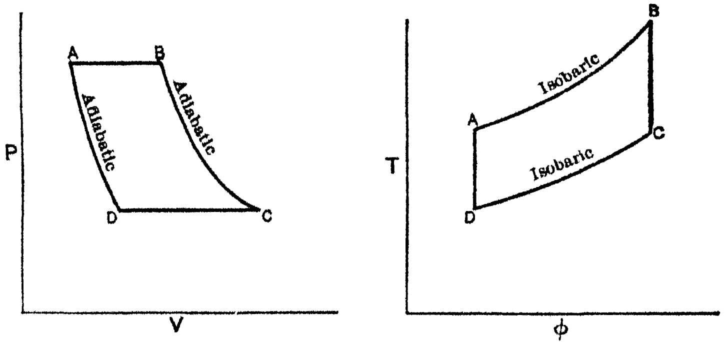
Work done $= C_v (T_B - T_A - T_C + T_D)$ B.t.u. (71)

$$\text{Efficiency} = \frac{C_v (T_B - T_A - T_C + T_D)}{C_v (T_B - T_A)} = \frac{T_B - T_A - T_C + T_D}{T_B - T_A} \qquad (72)$$

$$= 1 - \frac{T_D}{T_A} = 1 - \frac{1}{r^{\gamma - 1}}. \qquad (73)$$

In the development of these equations, C_v is assumed constant. The last form of equation (73) is derived by the aid of the adiabatic equation $Pv^{\gamma} =$ constant.

(*e*) *Brayton Cycle.* Figs. 255 and 256.



FIGS. 255, 256.—BRAYTON CYCLE.

Heat supplied per lb. of gas (along *AB*)

$$= \int_{T_A}^{T_B} C_p dT = C_p (T_B - T_A) \text{ B.t.u.} \qquad (74)$$

Heat discharged (along *CD*)

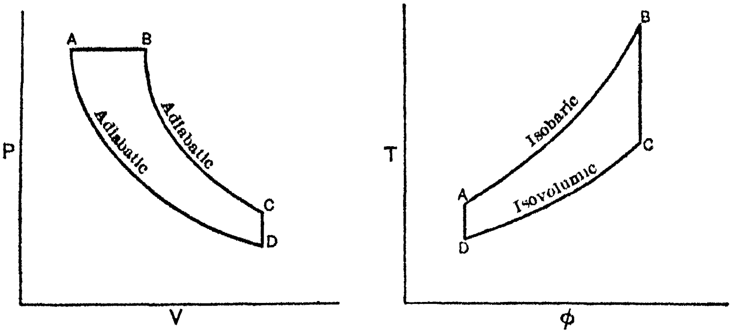
$$= \int_{T_D}^{T_C} C_p dT = C_p (T_C - T_D) \text{ B.t.u.} \qquad (75)$$

Work done $= C_p (T_B - T_A - T_C + T_D)$ B.t.u. (76)

Efficiency $= \dfrac{C_p (T_B - T_A - T_C + T_D)}{C_p (T_B - T_A)}$. (77)

This reduces to exactly the same form as equation (73). The assumption is again made that specific heat is constant.

(*f*) *Diesel Cycle* (*modified Brayton*). See Figs. 257 and 258.



FIGS. 257, 258.—DIESEL CYCLE.

Heat supplied per lb. of gas (along *AB*)

$$= \int_{T_A}^{T_B} C_p dT = C_p (T_B - T_A) \text{ B.t.u.} \qquad (78)$$

Heat discharged per lb. of gas (along *CD*)

$$= \int_{T_D}^{T_C} C_v dT = C_v (T_C - T_D) \text{ B.t.u.} \qquad (79)$$

Work done $= [C_p (T_B - T_A) - C_v (T_C - T_D)]$ B.t.u. (80)

$$\text{Efficiency} = \frac{C_p(T_B - T_A) - C_v(T_C - T_D)}{C_p(T_B - T_A)} = 1 - \frac{C_v(T_C - T_B)}{C_p(T_B - T_A)}$$

$$= 1 - \frac{T_C - T_D}{\gamma(T_B - T_A)}. \qquad (81)$$

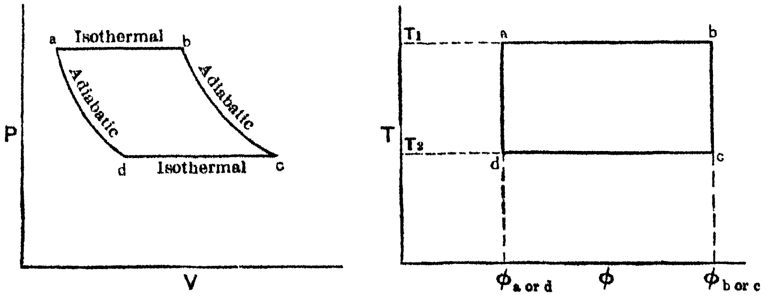
$$= 1 - \frac{\delta^\gamma - 1}{\gamma r^{\gamma-1}(\delta - 1)}. \qquad (82)$$

In the last equation

$$r = \frac{V_D}{V_A}, \quad \delta = \frac{V_B}{V_A}, \quad \text{and } \gamma = \frac{C_p}{C_v}.$$

Vapor Cycles. —The behavior of vapors when subject to cyclic variations under differing conditions of saturation or of superheat differs very much, depending mainly upon the nature of the vapor used. To discuss even a few of the vapors would lead to considerations somewhat beyond the scope of this book. Fortunately, in engineering practice we find only two, steam and ammonia vapor, used to any considerable extent, and the discussion will therefore be confined to these two vapors. The behavior of ammonia vapor is probably more easily understood in connection with machines in which it is used, and consequently the consideration of ammonia vapor (refrigerating) cycles will be taken up under refrigerating machines.

Cycles for Steam — *(a) Carnot Cycle for Saturated Steam.* Figs. 259 and 260.



FIGS. 259, 260.—CARNOT CYCLE FOR STEAM.

Heat supplied per lb. of vapor $= r_1(x_b - x_a) = T_1(\phi_b - \phi_a)$ B.t.u. (83)

The first expression of the above equation generally reduces to $x_b r_1$, because it is usually assumed that at a the material is liquid at the temperature corresponding to the pressure P_a. r_1 stands for the latent heat of vaporization at pressure $P_a = P_b$. The factor x, with proper subscript, expresses the quality of the vapor.

Heat discharged per lb. of vapor

$$= r_2\,(x_c - x_d) = T_2\,(\phi_c - \phi_d)\ \text{B.t.u.} \qquad (84)$$

Here r_2 is the latent heat of vaporization at the pressure $P_d = P_c$.

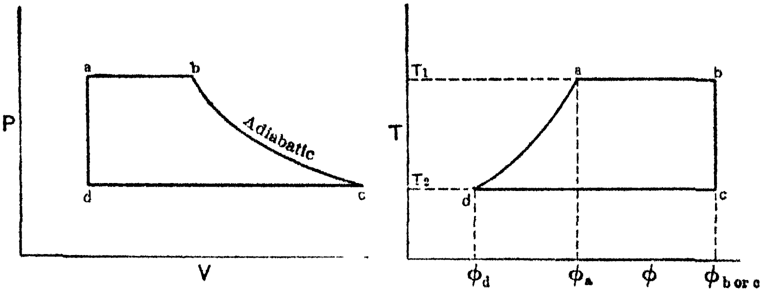
The work done is of course equal to the heat supplied less the heat discharged, and the efficiency may be written:

$$\text{Efficiency} = \frac{x_b r_1 - r_2 (x_c - x_d)}{x_b r_1} = \frac{(T_1 - T_2)\ (\phi_b - \phi_a)}{T_1\,(\phi_b - \phi_a)}$$

$$= 1 - \frac{r_2 (x_c - x_d)}{x_b r_1} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}. \qquad (85)$$

NOTE: *Carnot Cycle with Wet, Dry, or Superheated Vapor.* It must be evident from an examination of the derivation of the above efficiency formula that the efficiency depends only upon the shape of the Carnot cycle in the $T\phi$ diagram and not upon the properties of the working substance. Since the Carnot cycle must always be made up of two reversible isothermals crossed by two reversible adiabatics, it must always have the same shape when drawn to $T\phi$ coördinates irrespective of the properties of the working substance. Hence the efficiency must be $\frac{T_1 - T_2}{T_1}$ for all cases

(*b*) *Clausius Cycle with Saturated Steam.* Figs. 261 and 262.



FIGS. 261, 262.—CLAUSIUS CYCLE, SATURATED STEAM.

Heat supplied per lb. of steam

$$= q_1 - q_2 + x_b r_1 = C_l\,(T_1 - T_2) + T_1(\phi_b - \phi_a)\ \text{B.t.u.}, \qquad (86)$$

assuming that C_l is constant.

$$\text{Heat discharged} = x_c r_2 = T_2\,(\phi_c - \phi_d)\ \text{B.t.u.} \qquad (87)$$

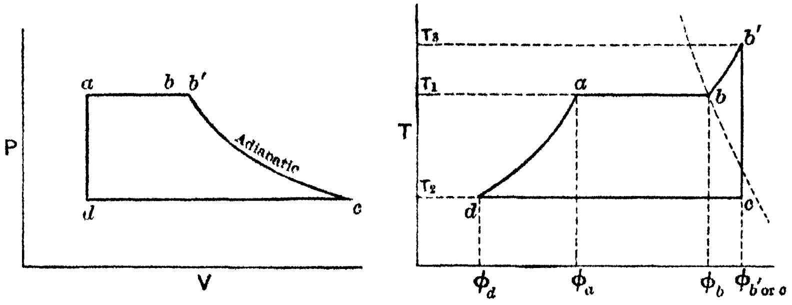
The work done is again the difference between equations (86) and (37), and the efficiency consequently is:

$$\text{Efficiency} = \frac{C_l(T_1 - T_2) + T_1(\phi_b - \phi_a) - T_2(\phi_c - \phi_d)}{C_l(T_1 - T_2) + T_1(\phi_b - \phi_a)} \tag{88}$$

$$= 1 - \frac{T_2(\phi_c - \phi_d)}{C_l(T_1 - T_2) + T_1(\phi_b - \phi_a)}. \tag{89}$$

It can be shown by inspection of areas in the $T\phi$ diagram that the term $\dfrac{T_2(\phi_c - \phi_d)}{C_l(T_1 - T_2) + T_1(\phi_b - \phi_a)}$ is greater than the term $\dfrac{T_2}{T_1}$, and hence that the efficiency of this cycle is less than the efficiency of the Carnot cycle between the same temperature limits.

(*c*) *Clausius Cycle with Superheated Steam.* Figs. 263 and 264.



FIGS. 263, 264. —CLAUSIUS CYCLE WITH SUPERHEATED STEAM.

Heat supplied

$$= q_1 - q_2 + r_1 + C_p(T_3 - T_1) \, . \tag{90}$$

$$= C_l(T_1 - T_2) + T_1(\phi_b - \phi_a) + C_p(T_3 - T_1) \text{ B.t.u.,} \tag{91}$$

again assuming that C_l is constant, and that C_p is constant.

$$\text{Heat discharged} = x_c r_2 = T_2(\phi_c - \phi_d) \text{ B.t.u.,} \tag{92}$$

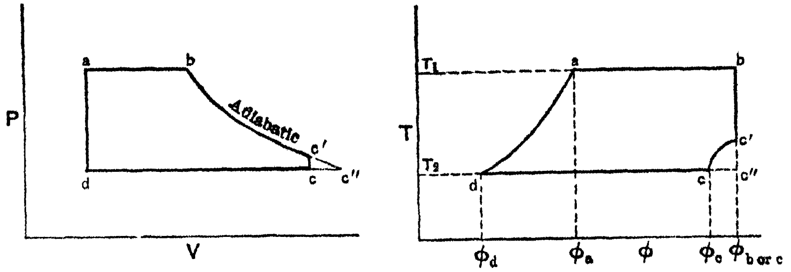
Work done = equation (91) − equation (92)

Efficiency

$$= \frac{C_l(T_1 - T_2) + T_1(\phi_b - \phi_a) + C_p(T_3 - T_1) - T_2(\phi_c - \phi_d)}{C_l(T_1 - T_2) + T_1(\phi_b - \phi_a) + C_p(T_3 - T_1)}$$

$$= 1 - \frac{T_2(\phi_c - \phi_d)}{C_l(T_1 - T_2) + T_1(\phi_b - \phi_a) + C_p(T_3 - T_1)}. \tag{93}$$

(*d*) ***Rankine Cycle with saturated steam.*** Figs. 265 and 266.



FIGS. 265-266.—RANKINE CYCLE WITH SATURATED STEAM.

Heat supplied

$$= q_1 - q_2 + x_b r_1 = C_l (T_1 - T_2) + T_1 (\phi_b - \phi_a) \text{ B.t.u.} \qquad (94)$$

Heat discharged (along $c'c$)

$$= (q_{c'} + x_{c'} \rho_{c'}) - (q_c + x_c \rho_c) = \overline{C}_v (T_{c'} - T_2). \qquad (95)$$

In equation (95) x_c and $x_{c'}$ are best determined from the $T\phi$ diagram. $\overline{C}_v$ is what might be called the mean constant volume specific heat of the wet steam in the temperature range $T_{c'}$ to T_2.

Heat discharged (along cd) $= x_c r_2 = T_2 (\phi_c - \phi_d)$ B.t.u. $\qquad$ (96)

The work done in the cycle is in this case equal to the sum of the quantities in equations (95) and (96) subtracted from the heat quantity in equation (94). The efficiency may be written:

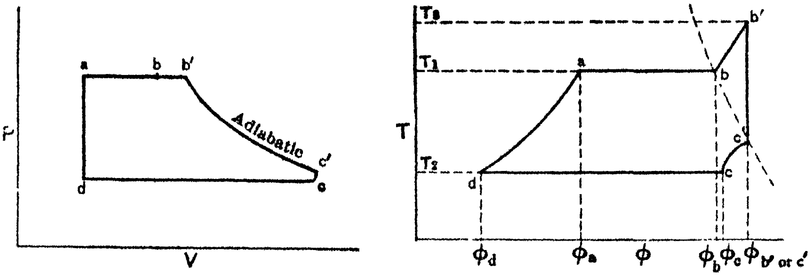
Efficiency

$$= \frac{C_l (T_1 - T_2) + T_1 (\phi_b - \phi_a) - \overline{C}_v (T_{c'} - T_c) - T_2 (\phi_c - \phi_d)}{C_l (T_1 - T_2) + T_1 (\phi_b - \phi_a)}$$

$$= 1 - \frac{\overline{C}_v (T_{c'} - T_c) + T_2 (\phi_c - \phi_d)}{C_l (T_1 - T_2) + T_1 (\phi_b - \phi_a)}. \qquad (97)$$

An inspection of the areas developed will show that the Rankine cycle is less efficient than the Clausius cycle, since the work equivalent to the area $c\ c'\ c''$ is not obtained in the former cycle.

(*e*) *Rankine Cycle with Superheated Steam.* Figs. 267 and 268.



FIGS. 267–268. — RANKINE CYCLE WITH SUPERHEATED STEAM.

Heat supplied

$$= q_1 - q_2 + r_1 + C_p(T_3 - T_1)$$

$$= C_l(T_1 - T_2) + T_1(\phi_b - \phi_a) + C_p(T_3 - T_1) \text{ B.t.u.} \qquad (98)$$

Heat discharged (along $c'c$) $= (q_{c'} + x_{c'}\rho_{c'}) - (q_c + x_c\rho_c)$

$$= \overline{C}_v(T_{c'} - T_2) \text{ B.t.u.} \qquad (99)$$

Heat discharged along $(cd) = x_c r_2 = T_2(\phi_c - \phi_d)$ B.t.u. (100)

Work done is again equal to the sum of the heat quantities in equations (99) and (100) subtracted from the quantity in equation (98), and the efficiency equation in the final form will be

$$\text{Efficiency} = 1 - \frac{\overline{C}_v(T_{c'} - T_2) + T_2(\phi_c - \phi_d)}{C_l(T_1 - T_2) + T_1(\phi_b - \phi_a) + C_p(T_3 - T_1)}. \qquad (101)$$

NOTE. In this cycle the point c', Fig. 268, is of course not necessarily on the steam curve, as indicated. The steam at release may be either superheated, dry or saturated, or wet. The second of these cases is represented in the figure.

CHAPTER XII.

THE MEASUREMENT OF LIQUIDS, GASES, AND VAPORS.

186. General Considerations. — The number of methods of measuring fluids is large. The choice of one or the other for any given case depends generally, first, upon the quantity of liquid to be measured, and second, upon the relative accuracy of the several methods that are available.

The methods in common use are by most authorities classed under two heads: *positive* and *inferential.* A positive method is one in which all of the liquid concerned in any given transaction is subjected to measurement, while by an inferential method only a certain part of the total quantity is measured. The proportion that this part bears to the whole must of course be known. Examples of positive methods are very common. To the class of inferential methods belong the method of Brauer for measuring water (see p. 367) and the use of so-called proportional gas meters for large capacities. Generally speaking, positive methods are more accurate than inferential, because they obviate the error involved in determining the proportionality of the part subjected to measurement. Such methods should therefore be preferred; unfortunately, however, the size of the apparatus or appliances required in many cases limits their applicability.

Another distinction that can be made between the various methods of measuring fluids is to class them as *direct* and *indirect.* The difference lies in the fact that in some cases it is possible to determine the entire quantity of any flowing fluid by direct measurement (weighing or volume determination). In other cases, since quantity is a function of velocity of flow and of stream cross section, total quantity may be found indirectly by determining each of these factors separately. In the majority of cases direct methods are more accurate than indirect, but, again, their field of application is limited to comparatively small rates of flow.

METHODS OF MEASURING WATER OR OTHER LIQUIDS.

187. Classification of Available Methods. — The various methods for measuring water may be enumerated in greater detail as follows:

I. Actual weighing. May be positive or inferential. When used positively this is the most accurate method known.

II. Measurement by means of orifices, such as weirs, nozzles, Venturi tubes, etc. The accuracy of these methods may be made almost anything by the care taken in arranging and calibrating the orifice. May be positive or inferential but generally used positively.

III. Measurement by volume displacement.

IV. Measurement by volume displacement in mechanical meters which are practically always hydraulic engines of some kind, being moved by all or a fraction of the liquid flowing through them. The motion is recorded automatically and the instrument is generally arranged to read cubic feet or pounds. Meters of different types have very different degrees of accuracy and even those of the same make and type often vary considerably among themselves and differ from time to time. Where great accuracy is required this method is only permissible when all precautions have been taken to determine the accuracy and constancy of the meter, or other vessel used to obtain volumes.

V. Measurement by determining the average velocity of flow in a channel or pipe of known cross section. This method is essentially inferential because it is impossible to obtain the velocity of all the material flowing, measurements being taken at different points in the section and averaged. It is often the only possible method, as in the measurement of the flow of large rivers, and for such work it must be considered satisfactory. The devices used for determining the velocity are floats, tachometers, Pitot tubes, hydrometric pendulums, current meters, etc.

I. MEASUREMENT BY WEIGHING.

188. Methods Used. — As already pointed out, weighing on calibrated scales is the most accurate method known for determining quantity of water. Its application is frequently limited by the failure of sufficient scale capacity. Where the quantities are greater than can be handled by the scales directly, recourse may be had to an indirect method of weighing by means of *calibrated tanks.* Such tanks, besides having the usual outlet in or near the bottom, should be furnished with an overflow pipe near the top to insure that the top level shall in all cases come the same. The best way to construct such an overflow pipe is to put the pipe through the walls of the tank, reaching to about the center, and to place an elbow on the end, which of course should turn up. This method of construction will make the top level come to the same height very quickly in all cases. Before use, the tanks are calibrated by weighing into them smaller quantities of water until they are filled to the top level. This method is to be preferred to computing capacity by the dimensions. During use it is then merely necessary to keep a record of the number of times each tank is filled and emptied. Since this method is fundamentally one of measurement by volume displacement it becomes necessary to record the *temperature* of the water used for calibration and of that used during a test, in order to make the necessary corrections for change of weight per unit volume should the two temperatures not be the same.

In computing the size and number of tanks required to handle any given flow of water, we must consider the capacity of the supply main and the shape, location, and size of the outlet opening. The supply main capacity, together with the size of the tank, determines the time required for filling, while the time of emptying depends upon the size of the tank and the conditions at the outlet.

The usual proceeding is to fix upon the approximate size of the tank, since that is often determined by questions of room, of manufacturing facilities, etc., and then to determine the necessary number of them by considering inlet and outlet conditions. In some cases of preparing for a test, tanks will be already on hand, and it then merely becomes a question of deciding whether they are large enough.

The question of determining the necessary size of supply main for a given desired capacity will be taken up in Art. 218. The time allowance (for weighing, opening, and closing valves, etc.) that should be made over and above the actual time required for filling from a given main, depends altogether upon the kind of tank installation, quality of help employed, etc., and its estimation must be left to the engineer. It might, however, be stated that it is not usually the filling but the emptying operation in which trouble may be encountered where large quantities of water are concerned.

The time of emptying any prismatic tank (usual case) may be computed from the formula:

$$T = \frac{2\,A\,(\sqrt{y_1} - \sqrt{y_2})}{CF\sqrt{2\,g}} \text{ seconds.} \qquad (1)$$

This applies to cases where the outlet is in the bottom, but can be applied with sufficient accuracy for approximate estimation in other cases.

A = the uniform cross section of the tank in square feet.

y_1 = water level at the beginning, and y_2 = water level at the end of the emptying operation. Both distances are measured in feet above a certain datum line, taken at the center of the outlet orifice, if that should be located in the side, or at the bottom of the tank if the outlet is located in the bottom. In any practical case the tank is usually emptied as completely as the location of the orifice permits, in which case $y_2 = 0$.

C = the coefficient of discharge of the outlet orifice.

F = the area of the orifice in square feet, and

$g = 32.2$.

The only factor in the above equation requiring judgment in assuming it properly is the coefficient C. Its value can be affected in a multiplicity of ways. Usually there will be a short nipple leading out of the tank, the outlet being controlled by a globe or gate valve. The nipple may be flush with the walls of the tank or project to a certain extent. Type of valve, length of pipe, and inside projection are all factors which affect C. Its value may be anything from about .55 to .97 or .98, depending mainly upon conditions at the entrance. For details see Art. 191. In general, a well-rounded (bell-mouth) orifice, flush with the inside walls and with a short tube leading to the valve, gives the most favorable conditions. A certain reduction, upon which there is, however, no definite data, should be made for the coefficient C that would normally apply to a given orifice for the effect due to valves. A gate valve is in this respect preferable to a globe valve, as it offers much less resistance to the flow.

Allowing the outlet pipe to project beyond the inside walls of the tank causes a decrease in the value of C. Sometimes, however, especially in the use of calibrated tanks, it is found desirable to locate the lower level quickly or very accurately, and it will be found that an elbow secured to the inner end of the outlet pipe, and turned either up or down, helps matters in this respect very much. Of course the value of C experiences a still further decrease by this proceeding.

The *arrangement of tanks* of course varies with the conditions of use. For the specific case of boiler supply it will in general be found best to have the weighing or calibrated tanks discharge into a tank on a lower level, from which the feed pump draws. A convenient arrangement for more than two tanks is to place them in an arc of a circle around the supply pipe and to have a single swing pipe which can be brought over each tank in turn.

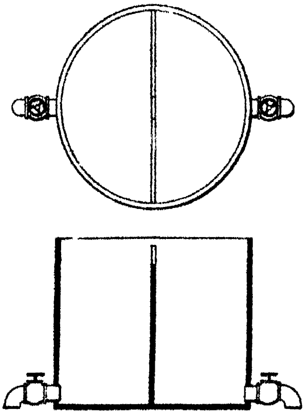


FIG. 269.

For small work a convenient form of calibrated tank used at Sibley College is illustrated in Fig. 269. The round tank is divided by a partition which has an overflow notch near the top. As one side fills the other side is emptied and the valve on that side closed. When the overflow begins, the swing supply pipe is shifted to the empty side, time is given for the overflow to cease, and the first side can then be emptied and made ready.

189. Sources of Error in Weighing. — For accurate work the scales should be calibrated. All valves, connections, and vessels should be free from leaks on the delivery side, as a considerable quantity of liquid can escape from what seems to be an insignificant opening.

Evaporation sometimes causes errors of considerable magnitude. With water at ordinary temperatures it may be neglected, but at a temperature above 150° F. it is best to keep all vessels covered. Some of the oils and alcohol may cause trouble in this way at ordinary temperatures and this possible source of error should always be recognized and guarded against when necessary.

II. MEASUREMENT BY USE OF ORIFICES.

The general term "orifices" is in this case intended to cover submerged orifices, nozzles, weirs, and Venturi tubes.

190. Submerged Orifices. — These orifices are in nearly all cases circular in cross section, and circular orifices will be the only ones here considered.

In the flow of water or other liquids the particles are urged onward by gravity, or an equivalent force, and move with the same velocity as bodies falling through a height equal to the head of liquid exerting the pressure. If this head in feet be represented by h, Fig. 270, and the corresponding velocity in feet per second by V, we have, neglecting friction losses,

$$V = \sqrt{2gh}. \qquad (2)$$

If we denote the area in square feet of the discharge orifice by F, the quantity discharged in cubic feet per second by Q, then, theoretically,

$$Q = VF = F\sqrt{2gh}. \qquad (3)$$

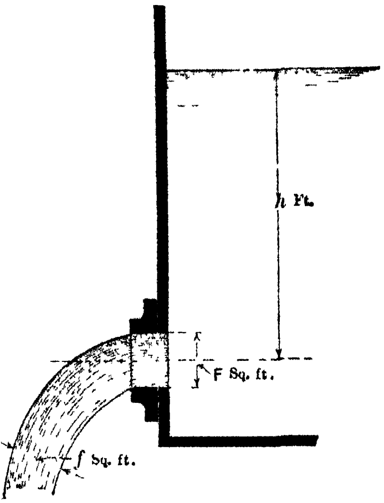


FIG 270.

It is found, however, in the actual discharge of liquid, that, except in rare cases: 1. the actual velocity of discharge is less than the theoretical; 2. the area of the stream discharged is different from the area of the orifice through which it passes (f different from F). These losses are corrected by introducing coefficients. The *coefficient of velocity* is the ratio of the actual to the theoretical velocity, and is represented by C_v. The *coefficient of contraction* is the ratio of the least area of cross section of the discharged stream to the area of orifice of discharge, and is denoted by C_c. The coefficient of *efflux* or *discharge* is the product of these two quantities, and is represented by C. The theoretical velocity V is attained most nearly at the minimum section of the stream in contact with air and the formulas are based upon the assumption that the maximum velocity is attained in this section. Hence the necessity of the contraction coefficient. The coefficient of velocity is very nearly unity for any smooth orifice and becomes of importance only when the liquid flows for a considerable distance over rough surfaces.

The actual discharge, then, becomes

$$Q_a = CQ = CVF = CF\sqrt{2gh} \text{ cu. ft. per sec.} \tag{4}$$

If V_a denotes the actual velocity of discharge, we shall have

$$V_a = C_v\sqrt{2gh}. \tag{5}$$

The coefficient C_v may be determined by experiment; it is nearly constant for different heads with well-formed simple orifices.

191. Value of the Coefficient C. — In practice it becomes necessary to determine the coefficient C by actual experiment, unless a standard calibrated orifice is available, according to which another for use can be made. For accurate work it is doubtful if even this proceeding is justified on account of the difficulty of duplication. Where the head under which the flow takes place varies widely, it further becomes necessary to determine C for various values of h, as the constant may vary slightly with the head for the same orifice.

As already pointed out, C for various orifices varies over a rather wide range, depending mainly upon conditions at entrance. The following figures show what values may be expected under the different conditions likely to be encountered.

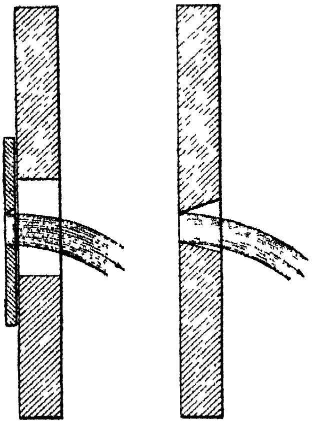


FIG. 271.

The constants for the orifice in thin plates (under *a*) are those compiled by F. Van Winkle in an article in *Power*,* while those under (*b*) are taken from Hütte and are originally due to Weisbach.

(*a*) Orifices in thin plates, sharp edges.

By this is meant either an orifice in a thin plate, or, if the wall is thicker, that the inner edges are so sharpened that no other part of the wall interferes in the stream filaments (see Fig. 271). This is probably the best type of standard orifice for water as long as the edges are unim-

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[*] *Power*, Aug. 25, 1908.

paired, because any wall effect is eliminated. The constants in the following tables apply to cases where there is complete contraction, and they may also be used for orifices in the bottom of vessels.

COEFFICIENTS OF DISCHARGE FOR ROUND VERTICAL ORIFICES IN THIN PLATE.

Head in Feet	Diameter of Orifice in Feet.						
	0.02	0.04	0.07	0.10	0.2	0.6	1.0
0.6	0.65	0.63	0.62	0.61	0.60		
0.8	0.65	0.63	0.62	0.61	0.60	0.59	
1.0	0.64	0.62	0.61	0.61	0.60	0.59	
2.0	0.63	0.61	0.61	0.60	0.60	0.60	0.60
4.0	0.62	0.61	0.60	0.60	0.60	0.60	0.60
6.0	0.62	0.61	0.60	0.60	0.60	0.60	0.60
20.0	0.60	0.60	0.60	0.60	0.60	0.60	0.59
100.0	0.59	0.59	0.59	0.59	0.59	0.59	0.59

COEFFICIENTS OF DISCHARGE FOR SQUARE VERTICAL OPENINGS IN THIN PLATE.

Head in Feet	Side of Square in Feet						
	0.02	0.04	0.07	0.1	0.2	0.6	1.0
0.6	0.66	0.64	0.62	0.62	0.61		
0.8	0.65	0.63	0.62	0.62	0.61	0.60	
1.0	0.65	0.63	0.62	0.61	0.61	0.60	0.60
2.0	0.64	0.62	0.61	0.61	0.61	0.60	0.60
4.0	0.63	0.61	0.61	0.61	0.61	0.60	0.60
6.0	0.62	0.61	0.61	0.61	0.60	0.60	0.60
20.0	0.61	0.60	0.60	0.60	0.60	0.60	0.60
100.0	0.60	0.60	0.60	0.60	0.60	0.60	0.60

COEFFICIENTS OF DISCHARGE FOR RECTANGULAR VERTICAL ORIFICES ONE FOOT WIDE IN THIN PLATE.

Head in Feet	Depth of Orifice in Feet						
	0.125	0.25	0.50	0.75	1.0	1.5	2.0
0.6	0.63	0.63	0.62	0.61			
0.8	0.63	0.63	0.62	0.61	0.61		
1.0	0.63	0.63	0.62	0.61	0.61	0.63	
2.0	0.63	0.63	0.62	0.61	0.61	0.62	0.63
4.0	0.62	0.62	0.61	0.61	0.61	0.61	0.62
6.0	0.62	0.62	0.61	0.60	0.60	0.61	0.61
10.0	0.61	0.60	0.60	0.60	0.60	0.60	0.60
				0.60	0.60	0.60	0.60

(*b*) External mouthpieces or ajutages, perpendicular to walls. Area F in formula always that corresponding to diameter d in the figures.

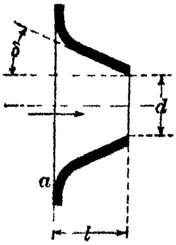

FIG. 272.

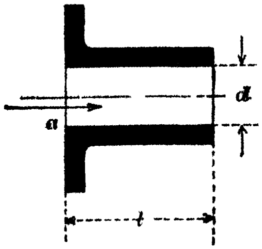

FIG. 273.

Shape of Fig. 272:

	Angle δ =	0°	$5\frac{3}{4}^\circ$	$11\frac{1}{4}^\circ$	$22\frac{1}{2}^\circ$	45°	$67\frac{1}{2}^\circ$	90°	
Edge *a* strongly chamfered,	C =	.97	.95	.92	.88	.75	.68	.63	(l = 3 d)
Edge *a* sharp,	C =	.83	.94	.92	.85				(l = 2.6 d)

Shape of Fig. 273:

	l =	1	2–3	12 d
Edge *a* sharp,	C =	.88	.82	.77

Edge *a* slightly rounded, $l = 3\ d,\quad C = .90.$

Edge *a* strongly rounded (bell mouth), $l = 3\ d,\quad C = .97.$

Shape of Fig. 272, except that flow is in opposite direction:

Angle δ =	0°	$22\frac{1}{2}^\circ$	45°	$67\frac{1}{2}^\circ$	90°	
C =	.54	.55	.58	.60	.63	(l = 3 d)

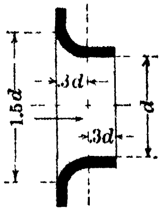

FIG. 274

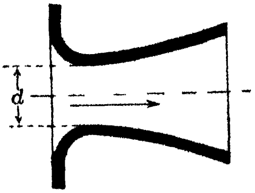

FIG. 275.

Shape of Fig. 274 (bell-mouth orifice): For $l = .6\ d$, $C = .96 - 1.00$, depending upon smoothness of walls.

Shape of Fig. 275: Depending upon length of piece and velocity of efflux, C may be from .96 to 1.5, referred to smallest cross section.

192. Method of Calibrating an Orifice. — The method of determining the constant C is a very simple operation. Means must be provided for weighing the liquid, while the rest of the work consists principally of determining the head h. The orifice must be carefully measured in order to determine F accurately. Runs should be made under various heads, and curves drawn between coefficient C and head h, to properly exhibit the relation.

Since accurate scales are a necessity for calibration, the question sometimes arises as to whether it would not be better to use scales throughout and to discard the orifice. The advantages of the use of an orifice over weighing are, however, manifold. An orifice, when properly cared for, needs to be calibrated only once, and the relation of constant C to head h can be used with confidence at any time thereafter. The labor involved in making measurements is much less than with weighing, since only one reading, that of the head, is required for each determination. Further, it is much easier to transport an orifice from place to place than it is to handle scales or meters. Finally, there is the possibility, by duplicating orifices and operating them in parallel, of measuring quantities of liquid far beyond ordinary scale capacity.

Regarding the latter point, care should be had to note that the several orifices are not too close together, since they may influence one another. The head over an orifice should be such that the surface of the water is not affected by the flow and there should be room around each orifice equivalent to at least the square of ten times the orifice diameter to prevent mutual interference.

The head on an orifice is measured to the center of the section of least area of orifice. The head h is always expressed in feet, since g in formula (4) is in feet. It may be measured directly, or, where the heads are high, it may be determined by means of a gauge or other manometer. In the latter case, pounds pressure per square inch or inches of mercury must be transposed to equivalent head in feet of water for substituting in the formula. For quick conversion, the following table may be of service:

TABLE SHOWING RELATION BETWEEN PRESSURE EXPRESSED IN POUNDS, AND THAT EXPRESSED IN INCHES OF MERCURY OR FEET OF WATER.

Pressure in Pounds per Sq. In.	70° F		
	Inches of Mercury.	Feet of Water.	Inches of Water.
1	2.0378	2.307	27.68
2	4.0756	4.614	55.36
3	6.1134	6.021	83.04
4	8.0512	9.23	110.72
5	10.1890	11.54	138.40
6	12.2268	13.85	166.08
7	14.2646	16.15	193.76
8	16.3024	18.46	221.44
9	18.3402	20.76	249.12
10	20.3781	23.07	276.80

Where any *other liquid than water* is concerned, and h is measured directly, equation (4) will give the correct result in cubic feet discharged per second. To convert this into weight requires a knowledge of the specific gravity of the liquid at the temperature of test. When the head is measured by means of a pressure gauge, *the head h to be used in the formula must be the height in feet of a column of the liquid concerned which will exert the pressure per square inch shown by the gauge.*

Most liquids other than water change specific gravity (weight per cubic foot) considerably with temperature, so that the influence of the temperature factor cannot always be left out of account. Even the specific gravity of water, contrary to a general assumption, varies enough to cause considerable error in some cases if temperature differences are neglected.

Thus, water at 40 degrees weighs 62.42 pounds per cubic foot; at 150 degrees its weight has decreased to 61.18 pounds per cubic foot. (See Kent, p. 688.)

193. Use of Orifices for Measuring Continuous Flow. — In practical work, orifices are not as much used for measuring continuous flow as the simplicity of the method would seem to warrant. The reason probably is that in one respect at least this simplicity is more apparent than real, that is, with reference to determining the head h. For widely varying rates of flow the determination of the average value of $\sqrt{h}$ is not easy, unless a device recording the variations continuously be employed. This fact at once restricts the use of the method to cases in which h is fairly constant. The proper arrangement of baffle plates to obtain still water around the orifice is important. (See Fig. 276.)

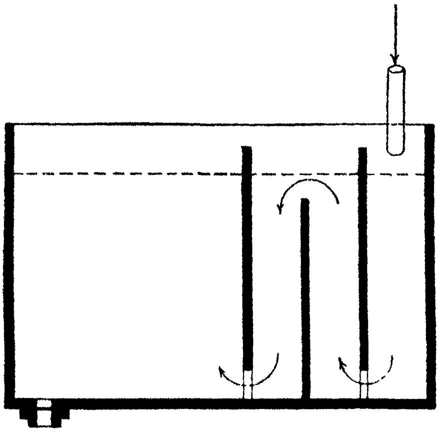

FIG. 276.

There is a method of orifice measurement, due to Brauer,* which avoids the necessity of noting the head h, and which is capable of being used for large capacities and varying rates of flow. Suppose that the vessel is fitted with two orifices of exactly the same proportions but of different diameters. Then the quantities of water flowing will be directly in the ratios of the areas of the two openings. This assumes that the constant C is the same for the two orifices for the same heads, which is true within the other errors of measurement. It then becomes necessary merely to weigh or otherwise determine the quantity of liquid flowing through the small opening, when the quantity flowing through the other is found by simply multiplying by the ratio of the areas. This scheme is capable of easy expansion of capacity by using several large orifices in parallel, the quantity of liquid subject to actual weighing remaining the same in all cases. The accuracy of the scheme depends directly upon the degree of accuracy with which the diameter of the small orifice is determined. The error made here can easily exceed allowable limits. Thus, if the small orifice is called .25 inch in diameter and is really only .24 inch, the area of the small orifice will be brought into the computation about 8 per cent too large, making the multiplier to obtain the quantity flowing from the orifice too small by about the same percentage. Errors of this magnitude are not usually permissible and the computation points out the necessity of accurately determining the diameter of the small orifice.

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* Zeitschrift des Vereins deutscher Ingenieure, 1892, p. 1493.

194. Measurement by Means of Nozzles. — The term *nozzle* usually refers to a conically convergent mouthpiece of the type shown in Fig. 272, screwed to the end of a hose or pipe. In the majority of cases, however, the length of the nozzle is greater than two to three times the smallest diameter, for which proportions the constants accompanying Fig. 272 are given. In other cases the nozzle may come within these proportions, and even an orifice in thin plate, when fastened by means of a cap to the end of a pipe, is commonly termed nozzle. From the foregoing it will be seen that the quantity of liquid discharged by a nozzle can be computed by equation (4); that is,

$$Q = CF\sqrt{2\,gh}. \tag{6}$$

The area F is the smallest cross section of the nozzle, in the ordinary type located at the extreme end. The head h is determined by a pressure gauge, the reading of the gauge being reduced to feet. Note that the head must be measured to the level of the smallest cross section of the nozzle. Thus, in Fig. 277 *a*, the distance h' is to be added to the head shown by the gauge, while in Fig. 277*b*, the distance h' should be subtracted to get the true head.

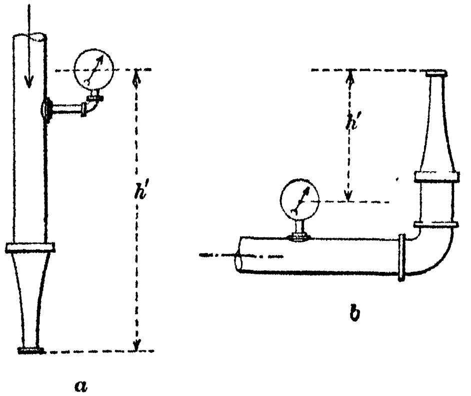


FIG. 277.

The coefficient of discharge C is again the product of the velocity coefficient C_v and the contraction coefficient C_c. In nozzles of some length as compared with the diameter at the end, the constant is likely to be considerably affected by the conditions of the walls. Since the constant also varies considerably with the angle of convergence, no general figures for the constant for such nozzles can be given and the only accurate method is to calibrate. The general directions given on page 365 for calibrating an orifice also apply in this case.

195. The Venturi Tube or Venturi Meter for Measuring Water or Other Liquid. — The Venturi tube is in principle nothing but an orifice placed in a pipe and through which the liquid to be measured is forced. The contracted passage increases the velocity of the liquid and causes a pressure loss, converted into velocity. This pressure loss may be measured, and since it is a function of the velocity, we are enabled to compute velocity and thus to compute the rate of flow through the pipe.

The Venturi tube as now built is shown in cross section in Fig. 278. It consists in reality of two convergent nozzles placed end to end with the smaller ends joined. The end through which the water enters is usually called the "upstream end," the other is designated the "downstream end," while the section of smallest diameter is known as the "throat." For measuring water it is desirable to have the throat area of such a size that the velocity through it shall be between the limits of 15 and 35 feet per second. The upstream end and the throat section are surrounded by pressure chambers, with which the respective sections communicate by means of a number of small openings, as shown. Gauges or other manometers connected to these chambers will then indicate the pressure heads at the sections, or by connecting the chambers to the two ends of a U-tube manometer, the reading of the latter will indicate the loss of pressure head between the upstream section and the throat.

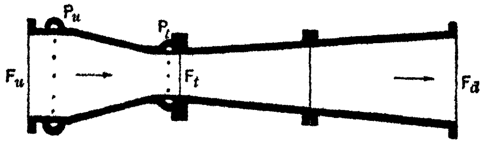


FIG. 278. — VENTURI TUBE.

The equation for the tube is best developed on the basis of Bernouilli's theorem, which states that *the total energy of a steadily flowing stream remains constant except for friction losses.*

In Fig. 279, let the pressure heads shown by the manometers at the up- and downstream section be H_1 and H_3, above a certain horizontal plane, as indicated. The downstream section is usually made equal to that upstream, so that H_3 will only differ from H_1 by the friction in passing through the tube, assuming the latter horizontal. The pressure at the throat may be anything, depending mainly upon the relation between the upstream and throat areas. If positive, it will be represented by some pressure head H_2; if negative (that is, if a vacuum), it will sustain a water column h_2. In that case the value of H_2 becomes $H - h_2$, in which H is measured as shown in Fig. 279.

Let the velocity in feet per second upstream be V_1, that at the throat be V_2, then according to the theorem cited, equating total

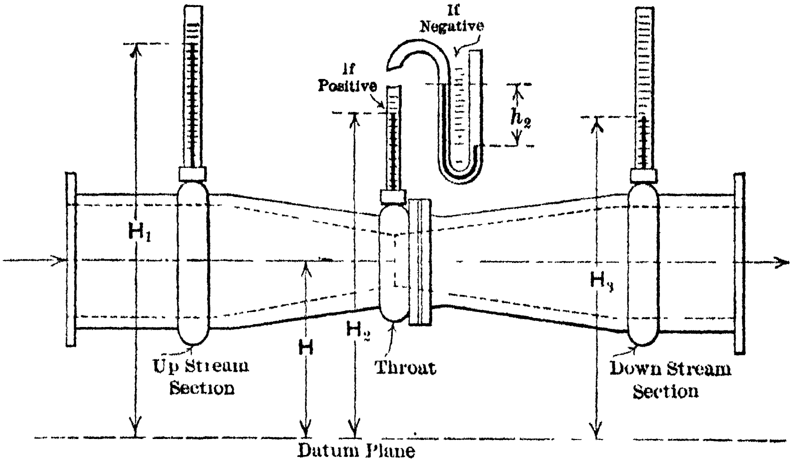


FIG. 279.

energies, we can write for the upstream and throat sections, assuming the tube horizontal, and the datum plane to coincide with the axis of the tube, so that $H = 0$,

$$H_1 + \frac{V_1^2}{2g} = H_2 + \frac{V_2^2}{2g} + H_f. \tag{7}$$

In this equation $\frac{V^2}{2g}$ represents the velocity head equivalent to a velocity V, while H_f is the head lost by friction between the upstream and throat sections. Note that all terms of the equation are expressed in feet, and that if the pressure heads are measured either by gauge or by mercury manometer the readings must be reduced to feet of water.

Herschel, in his experiments, showed that the loss of head H_f, due to friction, is very small in a properly made tube, and can be taken care of in the final formula by a coefficient. For the present, therefore, H_f can be neglected. Further, if F_u = the area in square feet of the upstream section, and F_t the area of the throat section, we must have

$$F_u V_1 = F_t V_2,$$

from which

$$V_1 = \frac{F_t}{F_u} V_2$$

Substituting this in equation (7) and neglecting H_f, we have

$$H_1 + \left(\frac{F_t}{F_u} V_2\right)^2 \cdot \frac{1}{2g} = H_2 + \frac{V_2^2}{2g}$$

or, solving for the throat velocity,

$$V_2 = \frac{F_u}{\sqrt{F_u^2 - F_t^2}} \sqrt{2g(H_1 - H_2)} \text{ ft. per sec.} \qquad (8)$$

Equation (8) is then modified by a coefficient (C) to adapt it to actual conditions.

The flow through the pipe may then be directly computed from

$$Q = CF_tV_2 = \frac{CF_tF_u}{\sqrt{F_u^2 - F_t^2}} \sqrt{2g(H_1 - H_2)} \text{ cu. ft. per sec.} \qquad (9)$$

In practice the pressure pipes leading from the two pressure chambers are connected to the two legs of a manometer, thus determining $(H_1 - H_2)$ directly. Note that the measuring liquid in the manometer must be heavier than water. Mercury is the liquid best adapted, but where small-pressure differences must be determined, either some multiplying form of manometer must be used, or a liquid only a little heavier than water, but which will not mix with water, can be employed. Note also that the connections leading to the manometer must be free from trapped air.

Herschel's experiments to determine the constant C gave values varying from .94 to 1.04. Out of fifty-five experiments, only four gave a value greater than 1.01, and only two a value less than .96. The tube experimented with had an upstream diameter of close to 1 foot, while the throat diameter was close to 4 inches. It seems from this that for ordinary work the coefficient C may be taken = 1.0. For more accurate work, of course, the tube requires calibration. The tube diameters should be carefully measured, as errors made with regard to them affect the result in the square.

$C = 1.00$ exactly if $\frac{1}{2}(H_1 - H_3)$ is deducted from $(H_1 - H_2)$, so accounting for the friction head H_f between upstream and throat sections.

This form of meter is very satisfactory and can be built to fit any commercial size of pipe. It is not subject to appreciable wear and cannot be easily damaged by any foreign substances carried by the liquid.

For best results the liquid must be free from mechanically entrained gas and must have fairly constant temperatures. It is possible to meter the liquids at widely differing temperatures by making allowance for changes in size of meter and density of liquid.

For comparatively brief tests and where the flow does not differ widely, the difference $(H_1 - H_2)$ may be noted at stated intervals. Where the pressure differences vary considerably it will be more accurate to find the average of the square roots of $(H_1 - H_2)$ than to find the square root of the average difference.

196. The Flow of Water over Weirs. — A weir, or weir notch, is primarily a dam or obstruction over which the water (or other liquid) is made to pass. The *head of water* producing the flow is the vertical distance to the surface of still water from the center of pressure of the issuing stream. The *depth of water* over the weir is measured vertically from the surface of still water upstream to the level of the bottom or sill of the notch. It should be noted that this depth is usually referred to as the head on the weir and is the factor generally designated by h or H in most weir formulas.

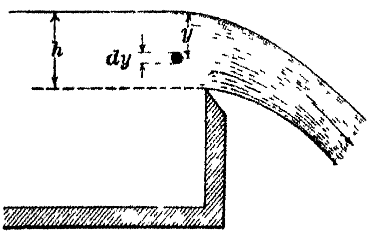


FIG. 280.

Rectangular Weir. Theoretical Equation. — Any small particle of water at a distance y, Fig. 280, below the surface of the stream has imparted to it a velocity $V = \sqrt{2gy}$. This is the velocity possessed by the narrow vertical plane of height dy and of length b, where b is the width of the weir notch. Hence the volume of water discharged through the infinitesimal orifice bdy will be

$$Q = bdyV = bdy\sqrt{2gy}. \qquad (10)$$

Integrating between the limits $y = 0$ and $y = h$, we will have

$$Q = b\sqrt{2g}\int_0^h y^{\frac{1}{2}}dy = \tfrac{2}{3}\, b\sqrt{2gh^3}\text{ cu. ft. per sec.} \qquad (11)$$

Note that in equation (11) b and h must be measured in feet.

Now it has been found that, due to various causes, the actual quantity of water discharged by a weir of given proportions is less than that found by equation (11). Hence the introduction of a coefficient C, and the practical form of the equation is then

$$Q = \tfrac{2}{3}\, Cb\sqrt{2\, gh^3}\text{ cu. ft. per sec.} \qquad (12)$$

This coefficient for small and accurate work should always be determined by calibration, but for larger work this proves in many cases impossible, and it will then have to be assumed upon the basis of other experiments. This makes it important to look into the causes influencing the value of C.

The causes operating to make C less than unity are primarily the following:

The depth of water (y) over the crest is less than the head h, measured to still water, see Fig. 281.

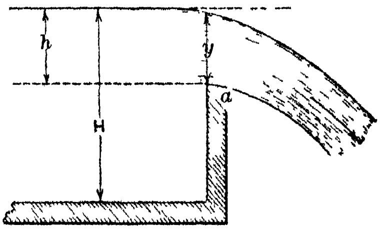
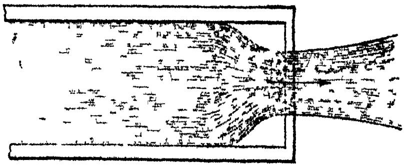


FIG 281

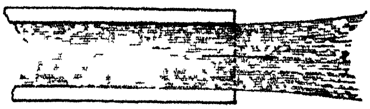
Contraction also takes place on the under side of the stream at a, if the stream is free (aired) underneath. If weir measurements are to be accurate and consistent, care should always be taken to see that the stream is free underneath after passing the notch, and does not in any way adhere.

Contraction of the stream sidewise (side or end contraction) may also occur, Fig. 282, which further decreases the actual flow, unless these contractions are suppressed, as in Fig. 283.



PLAN VIEW

FIG. 282 — SHOWING SIDE OR END CONTRACTION IN WEIRS



PLAN VIEW

FIG. 283 — WEIR WITH SIDE OR END CONTRACTION SUPPRESSED

Another factor strongly influencing the value of C is the velocity of approach. If the water has considerable velocity as it approaches the weir, C will be increased. This effect is connected with the relation between the total depth of the channel H and the head h, Fig. 281.

While the suppressing of end contraction may increase the value of C, it should also be remembered that for small weirs this suppression may bring the side walls so close that the friction of the water against them begins to affect the discharge. It must be evident from the foregoing that there is rather wide latitude in the choice of the value of C, and that all conditions of use of the notch must be definitely known before an intelligent choice can be made, if no calibration is possible.

From experiments made at Lowell, Mass., J. B. Francis concluded that the effect of end contraction could be taken into account by subtracting from b, the width of the crest, an amount equal to one-tenth of the head h over the weir. If n = the number of end contractions, his formula reads,

$$Q = \tfrac{2}{3} C \, (b - .1\, nh) \sqrt{2gh^3} \text{ cu. ft. per sec.} \qquad (13)$$

For the ordinary rectangular notch, since $n = 2$, this reduces to

$$Q = \tfrac{2}{3} C \, (b - .2\, h) \sqrt{2gh^3} = 5.3\, C \, (b - .2\, h) h^{\frac{3}{2}} \qquad (14)$$

Francis found the value of C in his experiments close to .62.

If there is considerable *velocity of approach*, it must be corrected for. This can be done on the consideration that the velocity head

$$h' = \frac{V'^2}{2\, g}. \qquad (15)$$

where V' = velocity of approach in ft. per sec., the value of which can be approximately determined at first by dividing the flow, as computed by ordinary formula, by the cross section of the weir channel.

Church ("Mechanics of Engineering," Art. 501) shows that taking into account the velocity of approach, but disregarding the contractions, the equation for the rectangular weir becomes

$$Q = \tfrac{2}{3} Cb \sqrt{2\, g} \left[(h + h')^{\frac{3}{2}} - h'^{\frac{3}{2}}\right] \text{cu. ft. per sec.} \qquad (16)$$

With the Francis correction for contraction, the final form of the equation then is,

$$Q = \frac{2}{3} C \sqrt{2g} (b - .1\, nh) \left[(h + h')^{\frac{3}{2}} - h'^{\frac{3}{2}}\right] \text{ cu. ft. per sec.} \quad (17)$$

This complete equation is little used; instead equation (14) is used, and changes are made in C to account for the effect of velocity of approach. See the discussion below regarding the value of C.

The variation of C has been studied by a number of experimenters, and the results of these made by Costel, Poncelet, Lesbros, Francis, Ftely-Stearns and by Frese were collaborated by the latter in the *Zeitschrift des Vereins deutscher Ingenieure*, 1890. Later work by Hansen has confirmed Frese's conclusions.* These were as follows:

For weirs without end contraction. —

$$C_1 = .615 + \frac{.0021}{h \text{ (meters)}} \text{ or } 0.615 + \frac{0.00064}{h \text{ (feet)}}.$$

This constant C_1 may be used directly in connection with equation (12). Where the velocity of approach is considerable, however, (say for small weir greater than $\frac{1}{2}$ ft. per sec.), C_1 must be multiplied by another factor $\varepsilon > 1$, to allow for the increased discharge. Frese found this factor to be

$$\varepsilon = 1 + .55\left(\frac{h}{H}\right)^2,$$

so that $$C = C_1\varepsilon.$$

For the significance of h and H, see Fig. 281.

Example: In a weir without end contraction, suppose $h = .50$ feet and $H = 2.5$ feet. Find C to be used in formula (12). Then

$$C_1 = .615 + \frac{.00064}{0.5} = .616, \text{ and } \varepsilon = 1 + .55\left(\frac{0.5}{2.5}\right)^2 = 1.022.$$

So that the coefficient C finally $= .616 \times 1.022 = .629$.

Weirs with Full Contraction. — This case is more complicated than the other because the degree of side contraction depends upon the relation between width of notch and total width of canal. Wherever possible it is therefore best, on the score of accuracy, to suppress the end contractions. Where this cannot be done, have the canal of such a size that both the velocity of approach and the influence of the side walls upon the contraction can be neglected. For such conditions Frese gives

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* Another paper containing an extended discussion of weir formulas is contained in "Water Supply and Irrigation" Paper No. 200, U. S. Geol. Survey (R. E. Horton). This touches upon the most important investigations, with the exception apparently of those of Frese.

$$C_1 = .576 + \frac{.017}{h\,(\text{meters}) + .18} - \frac{.075}{b\,(\text{meters}) + 1.2}$$

or

$$0.576 + \frac{0.0052}{h\,(\text{feet}) + 0.59} - \frac{0.0023.}{b\,(\text{feet}) + 3.9}\,.$$

This value of C_1 again applies to equation (12).

For the case where the canal has a width B and a depth H and neither the velocity of approach nor the effect of the walls can be neglected, the above coefficient C_1 must again be multiplied by a factor

$$\varepsilon = 1 + \left[.25\left(\frac{b}{B}\right)^2 + \varepsilon'\right]\left(\frac{h}{H}\right)^2$$

in which

$$\varepsilon' = .025 + \frac{.0375}{\left(\frac{h}{H}\right)^2 + .02}\,.$$

The coefficient for equation (12) will then again be

$$C = C_1\varepsilon.$$

Example: Assume $h = 1$ foot, $b = 5$ feet, $H = 4$ feet, and $B = 10$ feet. Find C

$$C_1 = .576 + \frac{0.0052}{1 + 0.59} - \frac{0.0023}{5 + 3.9} = .584$$

$$\varepsilon' = 0.025 + \frac{0.0375}{(\frac{1}{4})^2 + 0.02} = 0.479$$

$$\varepsilon = 1 + \left[.25\left(\frac{5}{10}\right)^2 + .479\right]\left(\frac{1}{4}\right)^2 = 1.034$$

$$C = C_1\varepsilon = .584 \times 1.034 = .604.$$

Weir notches having shapes other than the rectangular have sometimes been employed. The principal ones among these are the triangular notch and the trapezoidal (Cippoletti) notch.

Triangular Weir Notch. — The triangular weir has the advantage that the cross section of the stream is geometrically similar in form whatever the value of h. Hence the discharge coefficient is very regular in its variation and is more nearly constant than for other forms of weir. It has been found * that C for a triangular weir varies with h by the function $C = m + \frac{n}{\sqrt{h}}$, m and n depending upon the angle of the notch and upon conditions of approach. The triangular weir with an acute angle of opening is peculiarly well fitted for the measurement of small flows. For large values of θ (see Fig. 284), say about 135 degrees, researches on the value of C are as yet lacking.

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* Strickland, London Engineer, Oct. 28, 1910, p. 598.

In Fig. 284, θ is the angle of opening, b the width of the crest for any head h. As in the case of the rectangular weir, the velocity that any small particle of water has at a distance y below the surface is

$$V = \sqrt{2gy} \text{ ft. per sec.} \qquad (18)$$

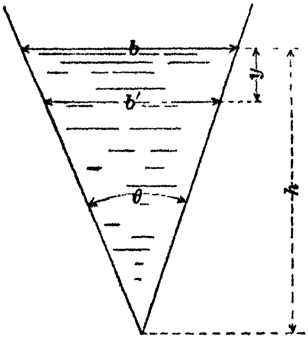

FIG. 284.

If b' represents the width of the notch at the head $h - y$, and δy the thickness of the filament of water at $h - y$, the quantity discharged will be

$$Q = b'\delta y \sqrt{2gy} \text{ cu. ft. per sec.} \qquad (19)$$

But $\dfrac{b}{h} = 2 \tan\dfrac{\theta}{2}$, from which $b = h\ 2 \tan\dfrac{\theta}{2}$, and similarly $b' = (h - y)\ 2 \tan\dfrac{\theta}{2}\cdot$ Hence

$$Q = 2 \tan\frac{\theta}{2}(h - y)\,\delta y \sqrt{2gy} = 2 \tan\frac{\theta}{2}\int_0^h (h - y)\sqrt{2gy}\,\delta y$$

$$= \left(\frac{8}{15}\sqrt{2g}\tan\frac{\theta}{2}\right)h^{\frac{5}{2}} \text{ cu. ft. per sec.} \qquad (20)$$

For practical use this equation is of course again adapted by introducing the coefficient C, so that

$$Q = C\left(\frac{8}{15}\sqrt{2g}\tan\frac{\theta}{2}\right)h^{\frac{5}{2}} \text{ cu. ft. per sec.} \qquad (21)$$

If the angle at the apex = 60 degrees,

$$\tan 30^\circ = .5773, \text{ and } Q = 2.46\,Ch^{\frac{5}{2}} \text{ cu. ft. per sec.} \qquad (22)$$

If the angle = 90 degrees,

$$\tan 45^\circ = 1.00, \text{ and } Q = 4.28\,Ch^{\frac{5}{2}} \text{ cu. ft. per sec.} \qquad (23)$$

Trapezoidal Notch. —To avoid the corrections for end contraction, Cippoletti in 1886 proposed the use of a trapezoidal notch of such dimensions that the area of the stream flowing through the triangular portion at the sides should be just sufficient to correct for the contraction of the stream in a rectangular weir. The proportions of such a weir, in terms of the length at bottom of the notch, are as follows: height equal to six-tenths the bottom length, width of top equal to the bottom plus one-fourth the height, added to each side; the tangent of the angle of inclination of the sides equal to 0.25. It is asserted that such a weir will give the discharge with an error less than one-half of one per cent. The formula for the use of such a notch would be simply

$$Q = \tfrac{2}{3}\, Cbh\,\sqrt{2\,gh} = 3.33\, bh^{\frac{3}{2}}, \qquad (24)$$

assuming that $C = .62$, and that the opening of the sides does truly compensate for variation of C.

197. Methods of Measuring the Head over Weirs. — The head in all cases is to be measured at a distance sufficiently back from the weir to insure a surface which is unaffected by the flow. The channel above the weir must be of sufficient depth and width to secure comparatively still water. The addition of baffle-plates, some near the surface and some near the bottom, under or over which the water must flow, or the introduction of screens of wire netting, serves to check the current to a great extent. Such an arrangement is sometimes called a tumbling-bay.

The object of the baffle-plates is to secure still water for the accurate measurement of the height of the surface above the sill of the weir. The same object can be accomplished by connecting a box or vessel to the water above the weir by a small pipe entering near the bottom of the vessel; the water will stand in this vessel at the same height as that above the weir, and will be disturbed but little by waves of eddies in the main channel. The height of water is then obtained from that in the vessel. Professor I. P. Church has the connecting-pipe pass over the top of the vessel and arranged so as to act as a siphon.

The accurate measurement of the head is of the greatest importance in weir measurements and careful work is necessary in all cases.

The Hook-gauge. — This consists of a sharp-pointed hook attached to a vernier scale, as shown in Fig. 285, in such a manner that the amount it is raised or lowered can be accurately measured. To use it, the hook is submerged, then slowly raised to break the surface. The correct height is the reading the instant the hook pierces the surface. To obtain the head of water flowing over the weir, set the point of the hook at the same level as the sill of the weir. The reading taken in this position will correspond to the *zero-head*, and is to be subtracted from all other readings to give the head of the water flowing over the weir.

In some forms of the hook-gauge the zero of the main scale can be adjusted to correspond to the zero-head, or level of the sill of the weir.

A type of hook-gauge used by Professor Frese has three points instead of one, all three being arranged in a right line. The instrument is so constructed and so set that two of the points pierce the water surface from below, as in the ordinary type, while the third (middle) one must touch the surface from above.

Floats. — Floats are sometimes used. They are made of hollow metallic vessels, or painted blocks of wood or cork, and carry a vertical stem; on the stem is an index-hand or pointer that moves over a fixed graduated scale.

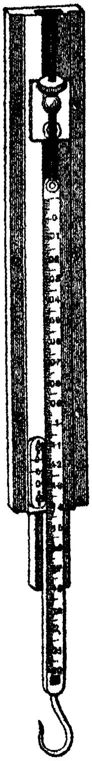

FIG. 285. HOOK-GAUGE.

198. Conditions Affecting the Accuracy of Weirs. — 1. The weir should be preceded by a straight channel of constant cross section, with its axis passing through the middle of the weir and perpendicular to it, of sufficient length to secure uniform velocity without internal agitation or eddies.

2. The opening itself must have a sharp edge on the upstream face, and the walls cut away so that the thickness shall not exceed one-tenth the depth of the overflow.

3. For complete contraction, the distance of the sill or bottom of the weir from the bottom of the canal should be at least three times the depth on the weir, and the ends of the sill should be at least twice the depth on the weir from the sides of the canal.

4. The length of the weir perpendicular to the current should be about three or four times the depth of the water.

5. The velocity of approach should be small; for small weirs

it should be less than 6 inches per second. Where this condition does not obtain, proper correction should be made.

6. The layer of falling water should be perfectly free from the walls below the weir, in order that air may freely circulate underneath.

7. The head on the weir should be measured with accuracy, at a point back from the weir unaffected by the suction of the flow and by the action of waves or winds.

8. The sill should be horizontal, the plane of the notch vertical.

199. Effect of Disturbing Causes and Error in Weir Measurements. — 1. Incorrect measurement of head. This may increase or decrease the computed flow, as the error is a positive or negative quantity.

2. Obliquity of weir; the effect of this or of eddies is to retard the flow.

3. Velocity of approach too great, sides and bottom too near the crest, contraction incomplete, crest not perfectly sharp, or water clinging to the outside of the weir, tend in each case to increase the discharge.

The causes tending to increase the discharge evidently outnumber those decreasing it, and are, all things being taken into account, more difficult to overcome.

200. Directions for Calibrating Weirs, Nozzles, and Venturi Meter. — This experiment combines the determination of the constant used in the flow formula for the three appliances mentioned. The water is sent through a Venturi meter to a horizontal header pipe of considerable capacity, into the bottom of which are several nozzles or orifices of varying construction, from each of which the water may be turned on or off as desired. The water discharges into a tumble-bay and flows over a weir without end contraction. As arranged in Sibley College, the water next reaches another tumble-bay at a lower level and is finally discharged over a rectangular notch with full contraction into tanks on scales.

The following are the directions:

Apparatus: Tanks and scales, hook-gauges for weirs, pressure gauge for nozzles, pressure gauges and manometer for Venturi tube.

1. Accurately level the sills of the weirs and see that the notches are in vertical planes.

2. Take the zero reading of the hook-gauges, by setting the point of the hook with a spirit level or other means at the level of the sill.

3. Turn off all of the cocks at the gauges; open valve above Venturi tube very slowly; open cocks at gauges; open valve to nozzle to be tested slowly until it is wide open.

In making a run, put a tare weight of —— pounds on the scales, then adjust the hook-gauges so the points just pierce the surface of the water. When the flow is steady (the level at the hook-gauge remaining constant), close the discharge valve of the weighing tank. Note the time to nearest second when the beam floats; then put —— pounds on the scale and note the time when the beam again floats.* The difference in time gives the number of seconds for the flow of —— pounds. At each different pressure on the Venturi meter the following observations are to be made:

a. Pressure at entrance to Venturi tube.
b. Pressure at throat of Venturi tube.
c. Pressure in receiver at nozzles.
d. Hook-gauge readings for each weir.
e. Weight of water discharged.
f. Time to nearest second.

In addition to the above runs at a graded series of pressures, the following runs are to be made:

g. Fully open all of the valves controlling the flow through the nozzles, then *slowly* open the valve at the entrance to the Venturi tube until a maximum difference is obtained between the Venturi tube gauges. Make a run of approximately —— pounds net of water, taking the same observations as before. This run is a maximum discharge through the Venturi.

h. For a maximum discharge over the weirs, in addition to all the nozzle valves being wide open, feed from a secondary supply pipe. Now very slowly open the valve at the Venturi tube until the water fills the weir notch; the water must not touch the upper edge of the notch. Make a run of —— pounds net, taking the usual observations.

The forms used for recording the observations and computed quantities follow.

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* Entrance of water to weighing system must be horizontal or nearly so for this method of weighing to be correct.

SIBLEY COLLEGE, CORNELL UNIVERSITY.—DEPARTMENT EXPERIMENTAL ENGINEERING.

CALIBRATION OF NOZZLES.

Type.................................
Date..............................
Observers.........................
Diameter.............................
....
....
Cross section........................
....

		Pressure on Nozzle		Weight of Water.					Nozzle Coefficient			Venturi Readings.			
No. of Run.	Duration in Seconds.	Pounds per Sq In	Feet of Water	Tare.	Gross.	Net.	Discharge, Cu. Ft. per Sec.	Velocity of Jet, Ft per Sec	Contraction	Velocity.	Discharge.	Gauge or Manometer A (Entrance)	Gauge or Manometer B (Contracted Section)	Total Difference in Head on Venturi (A)-(B)	Coefficient of Discharge.

SIBLEY COLLEGE, CORNELL UNIVERSITY.—DEPARTMENT EXPERIMENTAL ENGINEERING.

CALIBRATION OF WEIRS.

Form.................................
Date..............................
Length...............................
Observers.........................
....
Hook-gauge Zero......................
....

			Weight of Water						
No. of Run.	Duration, Seconds.	Hook-gauge.	Tare.	Gross.	Net.	Cu. Ft. per Sec.	Velocity of Approach (Ft. per Sec.)	Total Head on Weir.	Coefficient of Discharge.

III. MEASUREMENT OF WATER OR OTHER LIQUID BY VOLUME DISPLACEMENT.

201. Calibrated Tanks. — The method of their use and the precaution to be observed have already been discussed in Art. 188.

202. Tank Meters. — Within recent years several forms of apparatus have been developed which are best described as tank meters. They are all arranged automatically to fill and empty vessels of fixed size and to record the number of such operations.

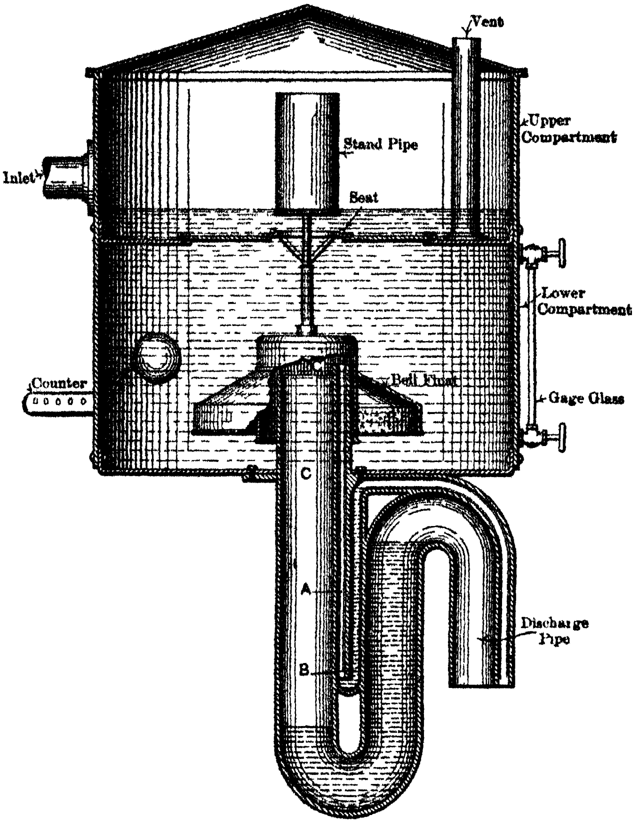

Fig. 286. — Wilcox Water Weigher.

The *Wilcox* water weigher, shown in Fig. 286, is a good example of this type. It is a tank divided into upper and lower compartments by a horizontal partition. The water enters the upper compartment, passes to the lower, in which it is measured, and then out through the U-shaped discharge pipe. The operation of filling and emptying the lower or measuring compartment is governed by the standpipe, bell-float and trip-pipe *AB* as follows:

The standpipe, which is open top and bottom and capable of vertical motion, is corrugated on the bottom and fitted to a seat in the horizontal diaphragm. When in its lowest position, water flowing into the upper compartment rises until it overflows into and through the standpipe, entering the lower chamber. After rising high enough in this chamber to seal the lower edge of the bell-float *C*, further rise compresses the air under the float and in the left-hand legs of the discharge pipe and the trip-pipe *AB*. This compression of air under the float immediately causes the latter to rise to its highest position, but as it is rigidly connected to the standpipe the latter is raised from its seat and the water in the upper compartment pours into the lower vessel. The compression of air under the float must continue until the pressure becomes great enough to break the seal in the trip-pipe. This action immediately reduces the pressure below the float, permits the latter to descend, sealing the upper chamber

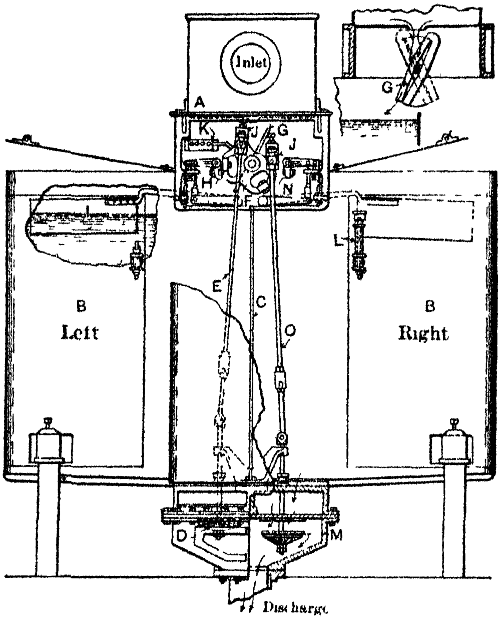

FIG. 287.—HAMMOND WATER WEIGHER.

against further discharge, and allows the water in the lower compartment to siphon out through the discharge pipe.

A mechanical counter records the total number of discharges, and the quantity of water measured is then found by multiplying the number of discharges by the quantity per discharge, the latter being a constant of the instrument.

This apparatus is built of material which will withstand the highest boiler feed temperatures, but it should be remembered when using it with water at high temperatures that the measurement is one of volume and hence correction for density is necessary.

Another apparatus of similar type is that known as the *Hammond* water weigher,* Fig. 287. It consists of two tanks side by side with common feed and discharge pipes. The feed is deflected into the one or the other by means of the deflector *G*, the position of which is controlled by that of the wrist plate *F*, the position of which in turn is controlled by the float *I* and the valve *M*. The float rising in the vessel being filled trips a latch which allows the wrist plate to rotate under the action of the head of water above the valve *M*, shifting the supply to the other tank and allowing the discharge of the tank just filled. An attached mechanical counter records the number of discharges as before.†

IV. MECHANICAL DISPLACEMENT OR POSITIVE METERS.

203. Water Meters.—The most complete discussion of recent years on water meters is probably that of Schonheyder before the Institution of Mechanical Engineers of Great Britain, 1900. He classifies all water meters as follows:

1. Low-pressure Meters.
2. Inferential Meters.
3. Volume or Capacity Meters.
4. Meters of the Venturi Class.
5. Waste Detection Meters.
6. Positive Meters.

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* *Power*, Nov. 24, 1908.

† Another illustration of this type of meter, much more complicated but also very accurate, will be found in the *Zeitschrift des Vereins deutscher Ingenieure*, Nov. 21, 1908.

A good water meter should meet the following requirements: (*a*) give accurate indications for widely varying rates of flow, from the smallest to the maximum suitable to the meter; (*b*) the indications should not change with time and should be independent of the water pressure; (*c*) the accuracy should not change with temperature; (*d*) the loss of pressure in passing the meter should be small; and finally, (*e*) the meter should require little attention and should not be especially vulnerable when metering dirty water.

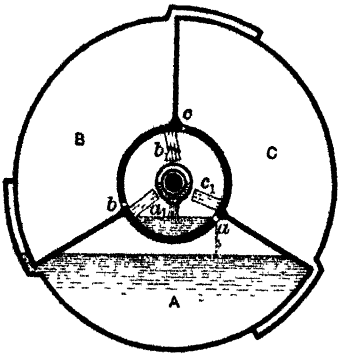

FIG. 288.—LOW-PRESSURE (OPEN-TYPE) METER.

1. *Low-pressure Meters.*—This type of meter is also sometimes called the open type. The latter designation implies that the entire main pressure is lost as the water passes the meter, which is the distinguishing characteristic of this class of meters. Fig. 288 illustrates one type, the Siemens meter.* The water supply coming through the central pipe flows into the central chamber, and from here flows through the openings *a*, *b* and *c*, into the respective compartments *A*, *B* or *C*, as the case may be. The lower compartment *A* is now filling, while *C* is emptying and *B* is out of action. When *A* is full, the supply will fill the central compartment, until the water flows through *b* into *B*, commencing to fill this. The left side of the drum thus gains in weight and at a certain level it will move 120 degrees in a counter-clockwise direction, after which *B* will be completely filled, and *A* starts emptying. Every complete revolution of the drum, therefore, discharges the volume contents of the three chambers. The short tubes, a_1, b_1 and c_1, serve to remove the air from the chambers.

Other types of low-pressure meter employ two vessels side by side, one emptying while the other is filling, and there are various other forms. The Wilcox and Hammond tank meters previously described also belong to this class of meter.

Low-pressure or open meters, when properly constructed, have the advantage of indicating accurately for widely varying rates of flow. They will take care accurately of the smallest dribble. Their use, however, requires attention, owing to the fact that they will meter water as fast as the inlet conditions will permit, irrespective of the demand. Thus, if the metered water is discharged into an intermediate receiver before use, and the demand on the receiver varies greatly, the meter must be watched or it may cause an overflow. This may be overcome in permanent installations by arranging a float control of the meter inlet.

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* Gramberg, *Technische Messungen*, p. 119.

2. *Inferential Meters.* —In this class the water is not actually measured, but the amount passing the meter is "inferred" from the number of revolutions made by a member which is caused to rotate by the current of water. To this class belong the so-called fan-wheel and turbine meters. These meters are often designated "velocity" or "current" meters, which is perhaps a more descriptive name, especially since the term "inferential" is also used for a method of measurement in which only a part of the liquid is actually measured.

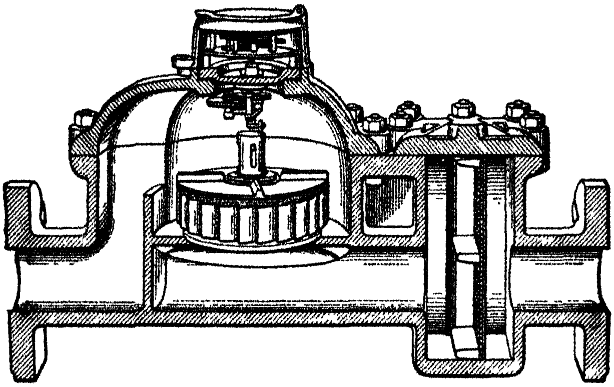

FIG. 289.—HERSEY TORRENT METER.

The number of different meters manufactured under this head is quite large, and it will therefore be possible to give only one or two typical examples of construction. The same remark applies practically also to all the other classes above enumerated.

Fig. 289 shows the Hersey torrent meter. Its operation is plain from the cut without much description. The water after passing the screen flows into the deflector, and then flows horizontally and practically radially through the wheel, causing it to revolve. The registering mechanism is of the ordinary type.

Another inferential meter is illustrated in Fig. 290. This shows the "Gem" meter, made by the National Meter Company. In this meter the wheel or propeller is made of spiral form. Water passing causes it to revolve, the number of turns being registered as shown.

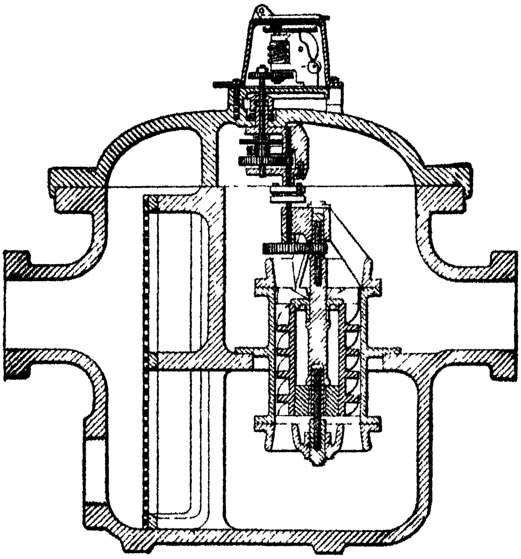

FIG. 290.—"GEM" INFERENTIAL METER.

The distinguishing feature of inferential meters is their large capacity as compared with meters of other types (see table following for some figures on the two meters illustrated).

Capacity of Current or Velocity Meters, cu. ft. per min.

Size, inches,	2	3	4	6	8	10	12
Normal capacity, cu. ft. per min.,	32	72	128	288	512	800	1152

Inferential meters are generally open to the objection that they will not register small flows, on account of the friction of the working parts, which requires a certain minimum velocity of water before any motion can take place. Consequently, for flows considerably below capacity, the registration is too small by from 2 to 5 per cent for the same reason. This feature is inherent in this type of meter, and the error can only be minimized by exact construction, light moving parts, etc. In the usual case it will be found that most of these meters will not register at all until the flow exceeds 2 per cent of the normal meter capacity. To overcome this difficulty two meters of this type, a large one and a small one, have sometimes been installed in parallel, so that the total quantity of water passing is obtained from the sum of the readings of the two meter dials. As long as the flow is small, only the small meter is working, the inlet to the large one being closed by an automatic valve. As the flow increases, the pressure difference between the two sides of the meters increases and at a certain value the valve to the large meter is opened. Thus this meter does not come into action until the region of its accurate registration is reached, and the error of the combination is only that incident to the non-registration of the small meter under flows which form only a very small percentage of the total normal capacity of both meters.

3. *Volume or Capacity Meters.* — There are a number of such meters made in this country. Among them may be mentioned the Crown, the Hersey, the Nash, the Keystone, and several other equally good ones. This type may be divided into three classes:

(*a*) Rotary meters.
(*b*) Disk meters.
(*c*) Piston meters.

Piston meters, although true volume meters, are generally known as positive meters, and are discussed in Class 5 below.

In *rotary meters* the moving member generally has motion in one plane only, and by this motion displaces, during one cycle, a definite volume of water, practically transferring it from the inlet to the discharge side. Differences in the construction of this moving member, usually called the piston, and differences in the cycle of its movement, constitute the main distinction between the rotary meters made by the different manufacturers. The principle of operation of these meters is rather simple, but the motion is complex enough to make it difficult of clear explanation, for which reason it is advised to make a study of a meter of this type and also the disk type.

The Hersey rotary meter, Fig. 291, is a representative type of this class. In this meter the piston *A* has six projections, while the cylinder *B* has six compartments. The motion of the piston is such as to produce motion in a circle of the pin *C* actuating the registering mechanism. The bottom head, *D*, contains two sets of openings leading to each compartment. Through one of these openings the water enters a compartment when the piston *A* by its motion clears it, while the other (the discharge) openings in this same compartment are closed by the piston. At the same time the discharge openings in the top head, *E*, which register with the discharge openings in the bottom head, are of course also held closed by the piston, there being no inlet opening in the top. Thus the compartment can fill. In the meantime the other compartments are either filling or emptying, and this results in a net water pressure tending to push the piston around its containing case or cylinder in a horizontal plane. When, in the compartment under consideration, the piston has cut off the supply opening, the discharge openings are freed, and discharge by displacement then occurs through the top and bottom discharge openings simultaneously, the bottom water finding its way to the central opening in the piston, through which it rises and joins the top discharge. The combined discharge

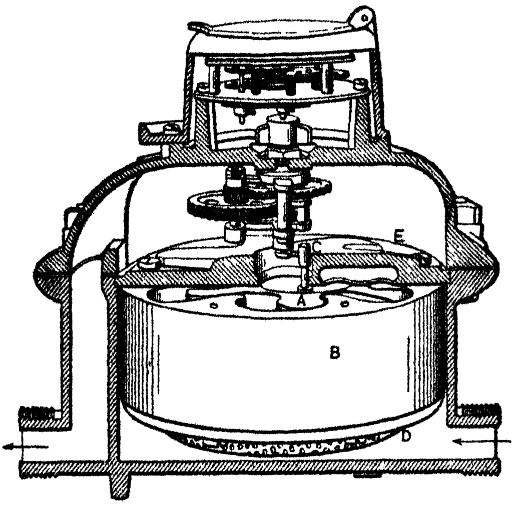

Fig. 291.—Hersey Rotary Meter.

then flows over the top head and out through the outlet at the left of the meter. In effect, therefore, the piston is nothing but a valve which, by the motion conferred upon it by the excess of pressure on one side over that on the other, controls inlet and outlet ports in the two heads.

A *disk meter*, the Nash, is shown in Fig. 292. A conical disk *A* moves in a chamber or compartment *B* (the floor of which is either a plane surface or only slightly conical and the roof of which is conical as shown) in such a manner as to produce motion in a circle of the pin *C* which moves the registering dial. The motion of the axis of the pin is peculiar and may best be compared with the motion of the axis of a spinning top when this axis does not coincide with a perpendicular through the contact point of the top. There is this difference, however; the meter disk has no motion of rotation about its own axis, for it is held in one position with respect to rotation by a vertical partition *D* which divides the compartment into two parts by passing from the side wall to the center, and the disk makes room

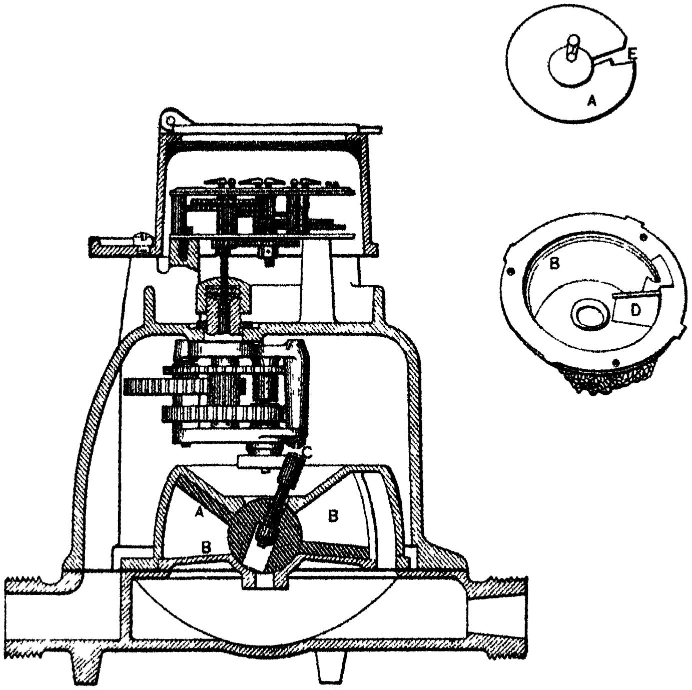

Fig. 292.—Nash Disk Meter.

for this partition by having cut into it a slot E from circumference to center (see detail figures of disk and compartment, Fig. 292). On one side of this partition in the side wall is located the water inlet port, while the outlet port is located on the other side of the partition. The latter port is shown in the cut. These ports are adjacent and only divided by the partition. Now in the position of the disk shown in the cut it will be noted that an element of each of the upper and lower surfaces is in contact respectively with the roof and floor of the compartment. The motion of the disk is such that some element of these surfaces will always be found in such contact, and hence the entire compartment is subdivided into four chambers, two on top of the disk and two below the disk, and two of these, one on top and one on the bottom, are open to the inlet port while the other two are discharging. The pressure difference between the filling and discharging sides causes the motion.

The advantages of rotary or disk meters consist in their simplicity of construction, light weight, small size, and comparatively low cost. They are open to the objection that no particular provision is made against leakage, and in many cases none is made for taking up wear. In spite of these facts, they are probably more extensively used in this country than any other type of meter.

The capacities of rotary or disk meters, irrespective of type or maker, seem to range about as follows:

Size of meter,	$\frac{3}{8}''$	$\frac{5}{8}''$	$\frac{3}{4}''$	$1''$	$1\frac{1}{2}''$	$2''$	$3''$	$4''$	$6''$
Greatest proper quantity per min., cu. ft.,	1	2	4	8	12	20	36	72	120

4. *Meters of the Venturi Class.* — The principle of the Venturi meter has already been explained (see Article 195). This meter has several marked advantages, as there pointed out, and has come into use in this country considerably, although mainly for waterworks use, where the quantity to be handled is large and where the proper amount of attention can be paid to the registering clockwork, which is in most cases rather complicated and costly. The last fact has probably militated against the more general adoption of this meter for small work.

5. *Waste Detection Meters.* — This is another special form of meter whose use is at present confined to waterworks and similar

plants. Schonheyder, in the paper mentioned, names only one of this class, the Deacon, shown in Fig. 293.* The water enters the upper end of the conoidal tube and acts upon a disk suspended centrally in the tube by a thin metallic cord passing through a gland. The greater the rate of flow, the greater the depression of the disk, whose position is continuously recorded by means of a pencil tracing a line on a paper moved by clockwork.

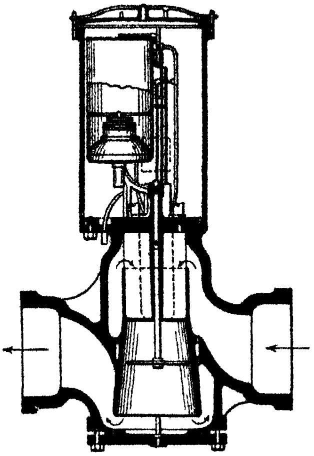

FIG. 293. — DEACON WASTE DETECTION METER.

6. *Positive Meters.* — These meters are generally piston meters, which "positively" measure the water by alternately filling and emptying. Most of the meters of this type now in use are duplex meters, that is, they have two pistons with their cylinders. These pistons may be connected to a common crank shaft, or they may operate like the steam pistons of the ordinary duplex pump, in which case each piston operates the valves for the other. It may be gathered from this description that these meters act very much like the ordinary piston pump reversed, and as a matter of fact they are in principle hydraulic power generators, the pressure of the water driving the pistons back and forth.

The piston meters in common use in this country are of the second class above mentioned, the pistons moving back and forth in the cylinders unconnected. It will suffice to show the construction of one of them. Fig. 294 gives an exterior (phantom) view of the Worthington piston meter, showing the interrelation of the main parts, while Figs. 295 and 296 give two cross sections. It will be seen that the pistons in their back and forth travel control ordinary slide valves operating on seats just above the bottom casting, which latter contains the inlet and outlet chambers. The plunger heads simply strike the valves near the end of the plunger travel and move them over at the proper time. One of the pistons works the registering mechanism by means of a lever and ratchet arrangement as shown. The end of the lever is moved by the plunger heads in the same manner in which the valves are operated.

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* Proc. Inst. Mech. Eng., 1900, p. 45.

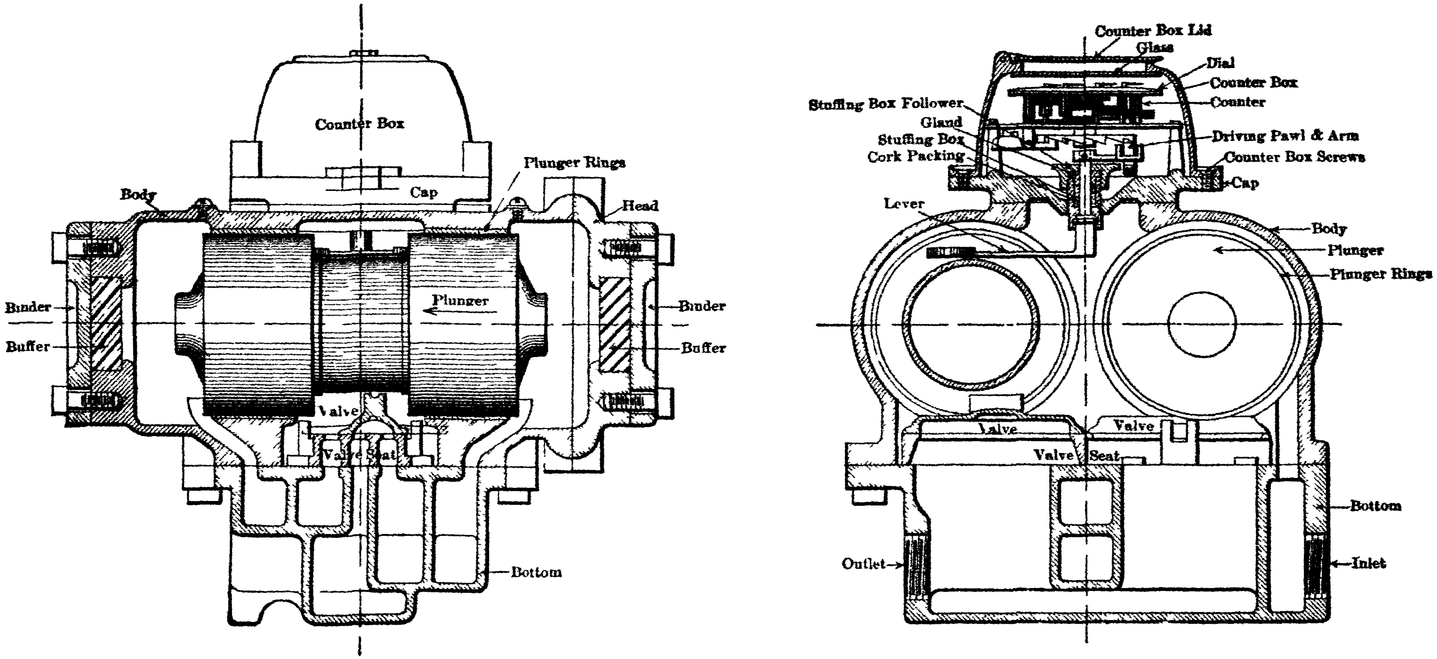
The capacity of piston meters can in general be assumed as follows:

Size of meter,	$\frac{5}{8}''$	$\frac{3}{4}''$	$1''$	$1\frac{1}{2}''$	$2''$	$3''$	$4''$	$6''$
Greatest proper quantity per min., cu. ft.,	$1\frac{1}{2}$	3	5	6	8	23	60	120

FIG. 294.—WORTHINGTON PISTON METER.

Meters of this type, when in good shape, are capable of giving accurate results. No provision, however, is usually made for taking up piston wear and leakage, and this materially affects the results after a period of service.

204. Hot-water Meters. — For ordinary service the running parts of most meters are made as light as possible and the material is often some hard-rubber composition. As long as the water is fairly cool, say not to exceed 110 or 120 degrees, these materials serve very well, but beyond this warping and twisting of the disks or pistons is apt to occur. Hence the ordinary type of meter cannot be used with hot water. For this service the meters are modified by making the running parts of some metal, usually some brass or bronze composition. Service with hot water is in many cases



FIGS 295 AND 296 —WORTHINGTON PISTON METER

quite severe on the meter, and it pays in general to install hot-water meters in a by-pass to prevent delays in the operation of a plant due to meter troubles.

205. The Calibration of Water Meters. — It should be taken as fundamental that no water meter of whatever make should be used without calibration, and not without frequent calibration if the work is of any importance. It is essential that the method of calibration should reproduce the conditions of use. It is consequently necessary in all cases where meters are used to note (*a*) the pressure, (*b*) the temperature, and (*c*) the rate of flow, in order to be able to reproduce these conditions when desired. The temperature reading is also necessary for the reduction of the meter readings, which are in cubic feet, to pounds.

In any test in which a meter is used it is usually best to install the meter in a by-pass. This is done for the double purpose of avoiding serious interruptions (as in boiler trials, for instance) and also to be able to calibrate the meter on the spot. For the method of doing this, see the meter connections recommended in the Code for Boiler Testing, Chapter XVII.

As a general rule, applicable to all meter calibrations, it may be stated that the greater the quantity of water drawn and weighed at the proper rate of flow, the better. The upper limit in this respect is generally soon reached on account of the limitations of tanks and scales, but for large meters it is almost necessary to provide capacity for not less than 20 to 30 cubic feet. It is generally more accurate not to draw a certain weight of water, but to operate the meter while the pointer on the first dial of the register moves from one definite mark on the dial to another definite mark. This avoids the estimation of fractional values on the dial between marks.

It sometimes happens that a meter is used on the same test under widely varying rates of flow and that the meter constant also varies greatly with the rate of flow. In that case a single calibration at the average rate of flow will not give the true correction factor, and it will be necessary for accuracy to determine the law of variation and to apply individual correction factors to various parts of the meter record.

V. MEASUREMENT OF THE QUANTITY OF LIQUID BY MEASUREMENT OF THE AVERAGE VELOCITY OF FLOW.

206. Measurements of this kind are made in closed pipes or conduits and in open channels. They are generally confined to cases where the flow of liquid to be measured is too great to be determined by the other methods so far discussed. The principle in each case is to determine the volume as the product of its two factors —cross section of stream and velocity of flow. Determination of the last factor only need here be considered.

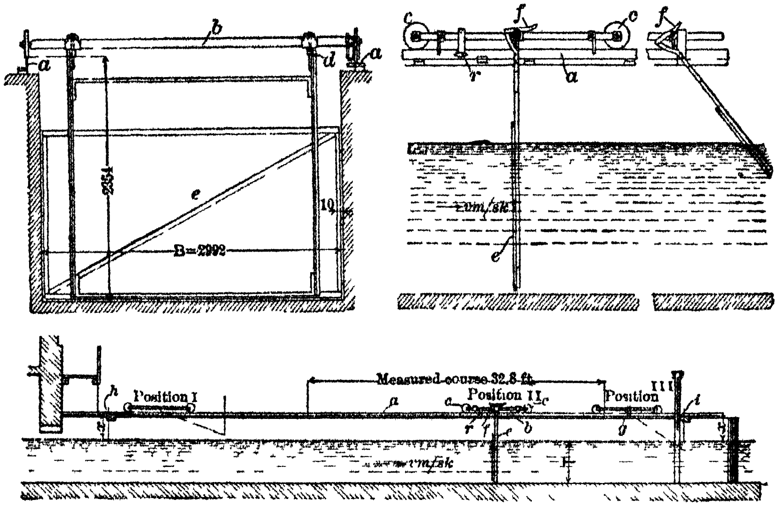
207. Measurement of Average Velocity in Open Channels and Streams. — The instruments and appliances used for this purpose come under three or four heads: floating bodies, or floats for short; pressure plates, hydro-dynamometers or hydrometric pendulums; tachometers or current meters; and the Pitot tube.

208. Floats. — For a complete discussion concerning the use of floats for measuring average stream velocity the reader is referred to almost any book treating on water-power development. The method is so little used in purely experimental engineering practice, being hardly accurate enough for efficiency computations, for instance, that it will suffice here briefly to point out the method.

The floating body should be small and have a density approximately that of water. Any small piece of wood or other material which will float and which can be observed from the banks may be used. If the stream is a natural one, as distinguished from the symmetrical channels often used for head or tail races in water-power plants, it is essential that the part of the stream chosen for the trial shall be fairly straight and as nearly as possible uniform in width and depth. The course may be from 50 to 200 feet long, a base line being laid off on the bank. At each end of the base line stretch a cord across the stream at right angles to the base line. These are the so-called transit lines. The transit lines are divided into any convenient number of equal divisions by means of tags fastened to the lines. Measuring the depth at each tag provides a means of drawing a profile of the stream for the up- and down-stream sections, thus determining mean cross section. The floats are released, one by one, some distance above the upper transit line, and time and spot of crossing the line are observed, the former by stop watch. The downstream observer similarly notes the place of crossing the lower transit line, and at the instant the float crosses gives a signal to the up-stream observer to note the time. In this way a number of velocities can be obtained across the width of the stream. If the stream is so wide that no cords can be stretched across, the method becomes more elaborate. In that case a theodolite is stationed at each end of the base line. When a float is started, the upper observer sights his instrument on the imaginary transit line and signals the time when the float passes. At the same time the lower observer focuses on the float and when the up-stream observer gives the signal, he reads the angle which locates the float as it crosses the upper transit line. Next the functions of the observers are reversed, the upper one following the float and noting the angle as it crosses the lower transit, the instant of which is signaled by the lower observer with his instrument clamped on the lower transit line. The time of transit has in the meantime been taken with a stop watch by a third observer.

Floats are of two kinds, —*surface or single floats* and *sub-surface* or *submerged* or *double floats.* The use of the latter is based on the consideration that a body simply floating on the surface of a stream may give a fairly accurate indication of the surface velocity, but tells nothing of the mean velocity over the entire cross section, which is certainly less than the surface velocity. Submerged floats are made by fastening a body capable of sinking to a surface float able to sustain the former by means of a light cord or wire. The latter is made of the proper length to maintain the bottom float where wanted, say about mid-depth. With these double floats it is assumed that the top float indicates much more nearly the average stream velocity. For very regular channels, *rod floats* reaching nearly to the bottom of the channel have often been used. They generally consist of a wood rod, weighted at the lower end so as to float nearly vertical. The method of using them is the same as in the other cases. For a discussion of the limits of accuracy of any of the float measurements, the reader is referred to Frizell's "Water Power" and Mead's "Water Power Engineering."

The latest development of measurement by means of floats is perhaps that reported in the *Zeitschrift des Vereins deutscher Ingenieure* for April 20, 1907. The method can be applied only in regular, artificial channels, but has been found to give very accurate results when used in the tail race of a turbine installation for determining the amount of water used. It consists of an "apron," constructed of a very light framework, over which varnished cloth is stretched. This apron *e* is supported on a carriage *c–c* (see Fig. 297), which runs on a track *a–a* the length of the canal. Accurate construction makes the force required to move the carriage very small. In the installation described, the canal had a width of about 10 feet, the depth of water was about 4.5 feet, while the length was 72 feet.



FIGS. 297 AND 298. — APRON METHOD OF DETERMINING VELOCITY IN A CHANNEL.

The carriage and apron weighed 88 pounds, but a force of only .88 pounds was required to move it. The method of using the apron is shown in Fig. 298. From the position I, at the left, the apron is released, and as the lower edge enters the flowing stream, the carriage is drawn forward. After a few feet of travel the apron hangs vertical, and is latched in this position by a special arrangement located at *f* on the carriage. The apron enters the water without shock and does not disturb the flow conditions appreciably. The measured course along the canal was 32.8 feet, and since this was passed through in some cases in about 12 seconds, it became necessary to use electrical instruments for recording time, distances, etc. The apparatus used is fully described in the article referred to. Fig. 298 shows the apron at position II as it is during the test. As soon as the end of the course is passed, the latch arrangement *f* is unhooked by contact with a rigid member, and the apron is then brought by the current into the inclined position shown at position III, which stops the carriage.

An analysis of this method of measurement shows that it is in principle identical with the ordinary float measurement except that it gives the average velocity over the entire cross section at a single trial. The writer of the above article showed that it could give very accurate results, even for very low speeds, down to .01 foot per second. But its application is likely to be restricted by the fact that straight channels of the requisite length are rarely available. It is also necessary that the containing walls shall be smooth and straight in order to allow of a close fit of the apron, to minimize the error due to water passing around the edges of the apron. For permanent testing flumes, the method probably deserves and will receive extended consideration.

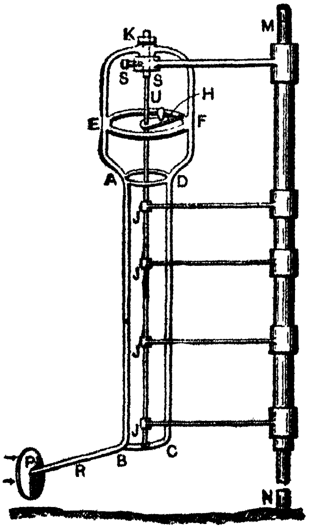


FIG. 299. — PERRODIL HYDRO-DYNAMOMETER.

209. Hydro-dynamometers and Hydrometric Pendulums. — The difficulty of obtaining satisfactory conditions for float measurements has led to the development of apparatus by which it is possible to obtain the mean stream velocity at a single well-defined section.* In pressure plates, or hydro-dynamometers, the current is made to impinge upon a plate, and the magnitude of the force or pressure exerted by the stream is then measured in various ways. To illustrate one type of this class, Fig. 299 gives an idea of the construction of the Perrodil hydrodynamometer as shown by Van Winkle. The pressure plate *P* is carried by the framework *ABCD*, which is supported at *K* by the standard *MN*, as shown. *UJ* is a torsion rod suspended at *K* on the upper end and fastened to the frame *ABCD* between *B* and *C*, as shown. *EF* is a graduated horizontal circle over which moves a pointer *H* which is rigidly fastened to the torsion rod. The instrument is used as follows: by loosening set screw *S*, the plate and framework are allowed to swing into the current like a loose rudder until the position of *P* is parallel to the current and the pointer indicates the zero on the scale *EF*. Next the pointer *H* is forced around, twisting the rod *UJ* until the plate *P* is perpendicular to the current. *S* is then fastened and the angle of torsion, which is a function of the pressure on the plate, may then be read.

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* See an article by F. Van Winkle on "Stream Flow at a Single Cross-Section," in *Power* for Aug. 30, 1910. The two books named in Article 208 also contain discussions on current meters and similar instruments.

The principle of the hydrometric pendulum is similar. This instrument consists of a ball, two or three inches in diameter, attached to a string. The ball is suspended in the water and carried downward by the current; the angle of deviation from a vertical may be measured by a graduated arc supported so that the initial or zero-point is in a vertical line through the point of suspension. If the current is less than 4 feet per second an ivory ball can be used, but for greater velocities an iron ball will be required. The instrument cannot give accurate determinations, because of the fluctuations of the ball and consequent variations in the angle. The formulæ for use are as follows: Let G equal the weight of the ball, D equal the weight of an equal volume of water; then $G - D$ is the resultant vertical force. Let F equal area of cross section of the ball, v the velocity of the current, c a coefficient to be determined by experiment; then we have the horizontal force $P = cFv^2$. Let angle of deviation be δ; then

$$\tan \delta = \frac{P}{G - D} = \frac{cFv^2}{G - D},$$

from which

$$v = \sqrt{\frac{(G - D) \tan \delta}{cF}}. \tag{25}$$

The best results with this instrument will be only approximations.

The one advantage that this instrument and the hydro-dynamometer have over current meters lies perhaps in the fact that the former can be used for lower velocities of flow than the latter. On the other hand, fluctuating velocity of current makes the determination of the mean reading very difficult with hydro-dynamometers.

210. Tachometers or Current Meters. — These instruments consist of light fan or propeller wheels, mounted either horizontally or vertically, and geared to a registering mechanism which indicates the number of turns made by the wheel in a given time, actuated by the current. The registering mechanism may be simply the ordinary type of revolution counter directly driven from the wheel shaft by worm and gear. The counter can be worked by lever and cord. The disadvantage of this scheme is that the counter is exposed to the grit and dirt that the water may carry, and further, that the instrument has to be bodily removed from the water to obtain its reading.

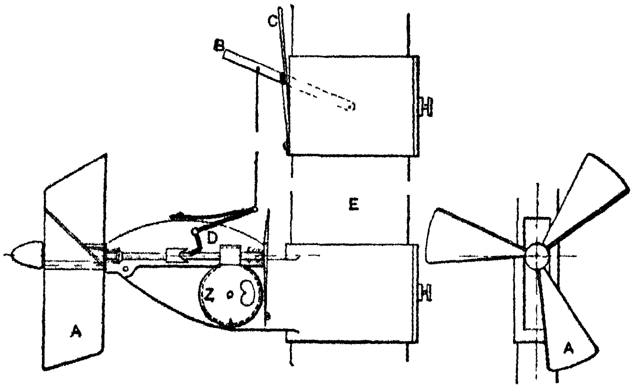

FIG. 300. — ERTEL CURRENT METER.

Attempts to avoid this have been made by gearing the wheel shaft to a light vertical shaft and having this drive the register above water. This scheme is generally bad, as it increases the friction seriously and thus raises the lower velocity limit below which the instrument will not record. The best solution of the problem is perhaps the electric recorder. In that case the friction loss is reduced to a minimum, since the wheel shaft is simply required to make and break electric contact. Another advantage of this arrangement is that the recorder may be located at any convenient distance from the point of observation.

Another good scheme for reducing friction is to use the so-called acoustic method of recording. In this scheme parts of the wheel shaft are arranged so as to give a distinct click at every turn. The sound may be transmitted to the operator by means of flexible tubing connected to the tubing supporting the current meter and through a simple form of receiver.

Current meters are generally provided with rudders to keep them parallel to the current, and, if readings at considerable depth must be taken, are also weighted so as to keep taut the rope by which they are suspended in the stream. For shallow channels rigid rod suspensions are preferred.

Fig. 300 shows Ertel's current meter* of the type first discussed. The instrument is supported by the rod *E*. *A* is the wheel and *Z* the counter. *B*-*C* is the cord and lever mechanism fastened to the top of the rod. Raising lever *B* releases the catch *D* and starts the counter. Pressing against *C* releases *B* and the flat spring on *D* then forces the catch back into position.

The next illustration, Fig. 301, shows the so-called Woltman mill, a type of current meter very largely employed. In this case the revolutions are recorded electrically. Instruments built on this principle are made by several manufacturers, differing in wheel construction and in method of recording.

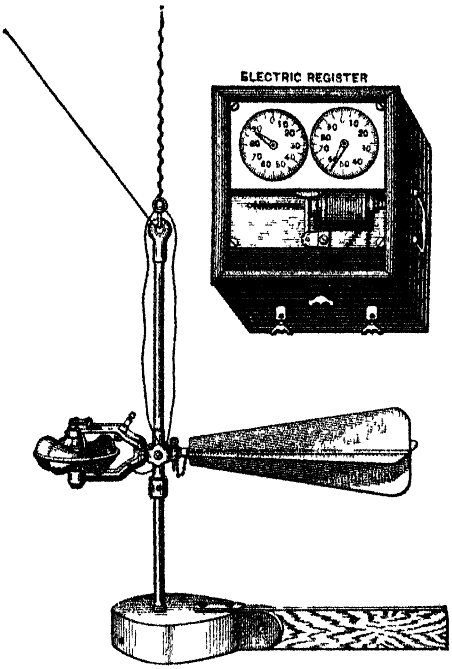

FIG. 301.—WOLTMAN MILL.

Finally, Fig. 302† shows one type of acoustic meter, *T* being the flexible transmitting tube and *R* the receiver.

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* Gramberg, *Technische Messungen*, p. 65.

† F. Van Winkle in *Power*, Aug. 30, 1910. See also Water Supply and Irrigation Paper, Nos. 94 and 95, U. S. Geological Survey, for further information regarding stream measurements and the meters used.

211. Calibration of Current Meters and Methods of Determining Average Velocity in Cross Sections. — This class of instruments is capable of giving very accurate results, but it requires rating or calibration. This may be done either by moving the instrument through still water at a definite velocity or by comparing the readings or the instrument with those of a standard instrument run alongside. The former method is perhaps the best, but not many testing flumes are available for the work of calibration.

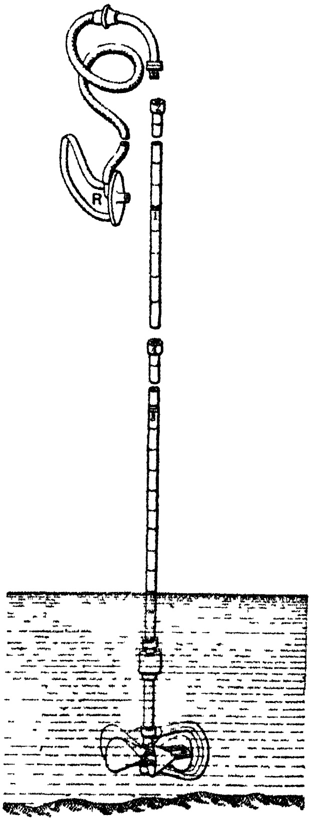


FIG. 302.—ACOUSTIC CURRENT METER.

Theoretically considered, the equation for a current meter should be $V = an$, where V = velocity in unit time, n = number of revolutions in the same time, and a = a constant peculiar to the particular type of wheel used. In practice, however, there is a certain current velocity, b, below which the wheel will not revolve on account of friction, and the real equation therefore is $V = an + b$. This is a straight-line law and the numerical constants of the equation may therefore be found from any two calibration tests made at different speeds. The manufacturers will ordinarily furnish calibration tables, giving n and V in parallel columns, but it is needless to say that for important work with an instrument that has been in use, recalibration is required.

Concerning the method of obtaining mean velocity over a given section, Van Winkle gives the following ways in which the current meter may be used: The mean velocity may be obtained either by taking the mean of averages of velocities at equally spaced points in this section; or by taking the mean velocity of the averages of velocities taken along equally spaced vertical lines; or by taking a single reading from the movement of the instrument uniformly over the section. The first and second methods are most appropriate for taking measurements with an instrument which is supported by a pole resting on the bottom of the stream, but for employment of instruments capable of being suspended, the last mentioned method has advantage in greater convenience and saving of time.

212. The Pitot Tube. — In its simplest form this is a bent tube held in the water in such a manner that the lower part is horizontal and opposed to the motion of the current in an open channel. By the impulse of the current a column of the water will be forced into the tube and held above the level of the water in the stream; this rise *DE* (see Fig. 303) depends upon the velocity of the water. If the height *DE* in feet above the surface of the water equal *h* and the velocity of the water equal *V* feet per second, we have

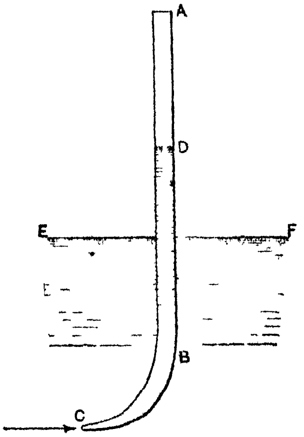


FIG. 303 — FUNDAMENTAL FORM OF PITOT TUBE

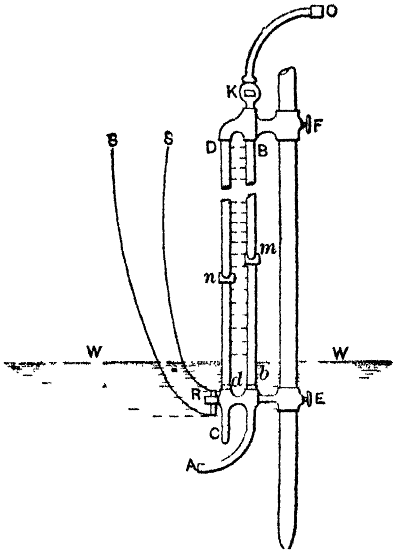


FIG. 304 — PITOT TUBE FOR OPEN CHANNEL WORK

$$V = C\sqrt{2gh} \qquad (26)$$

in which *C* is a coefficient to be determined by experiment. Concerning the value of *C*, see Art. 214.

The Pitot tube, as ordinarily used for open channel work, consists of two tubes, one, *AB*, bent, as in Fig. 304, the other, *CD*, vertical.

The mouth-pieces of both tubes are slightly convergent, to prevent rapid fluctuation in the tubes. These tubes are so arranged that both can be closed at any instant by pulling on the cord *ss* leading to the cock *R*. Between the glass tubes *dD* and *bB* is a scale which can be read closely by means of the sliding verniers *m* and *n*. The tubes are connected at the top, and a rubber tube with a mouth-piece *O* is attached.

In using the instrument it is fastened to a stake or post by the thumb-screws *EF*; the bent tube is placed to oppose the current of water, the cocks *K* and *R* opened. The difference in height of the water in the tubes will be that due to the velocity of the current. The water in the column *dD* will not rise above the surface of the surrounding water, and the instrument may be inconvenient to read. In that case some of the air may be sucked out at the mouth-piece *O*, and the cock *K* closed; this will raise the water in both columns without changing the difference of level, so that the readings can be taken in a more convenient position; or by closing the cock *R*, by pulling on the strings *ss*, the instrument may be withdrawn, and the readings taken at any convenient place.

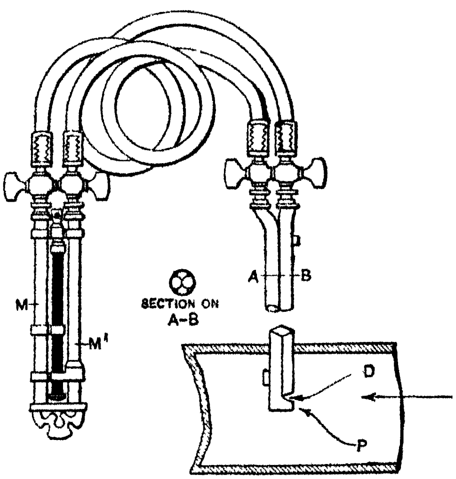


FIG. 305.—PITOT TUBE FOR CLOSED CONDUIT.

213. **Measurement of Average Velocity in Closed Pipes and Conduits.** — The instrument best adapted to this purpose is a modification of the Pitot tube (see Fig. 305). Here *P* is the impact opening facing the current, while *D* is the static opening at right angles to the flow. Both tubes are connected by flexible hose to the differential manometer *MM'*. Since the opening *P* registers both the static and the velocity head, while *D* registers the static head only, the static head cancels out and *MM'* shows only the velocity head *h*, from which $V = C\sqrt{2gh}$, as before. Hence the reading of the

manometer is independent of the static pressure in the pipe, but for accurate work it is absolutely essential that *D* register the static head properly.

214. Conditions Affecting the Value of the Constant C in the Best Form of Pitot Tube. — There has been considerable controversy concerning the equation of the Pitot tube, and some authorities have given the theoretical equation $V = \sqrt{gh}$ instead of $V = \sqrt{2\,gh}$. The work of later experimenters, however, has shown that the latter formula is the correct one, and also that with proper construction of tube and proper handling the constant *C* may with confidence be assumed = 1.0. For a complete discussion of this matter see the following papers: Vol. XXII, Trans. A.S.M.E., p. 284; Mr. W. M. White, in the Journal of the Association of Engineering Societies, August, 1901; Prof. W. B. Gregory, Vol. XXV, Trans. A.S.M.E., p. 184; Prof. W. B. Gregory and E. W. Schoder, Vol. XXX, Trans. A.S.M.E., p. 351.

It would appear that where the constant *C* is not practically 1.0 in the equation, the trouble may be found either at the impact or at the static opening. To get a true impact indication seems, however, to be a simple matter, it being apparently merely necessary to have a small hole, say not over $\frac{1}{4}$ inch, and better smaller, facing accurately against the current, and to have the shape of the tube in the immediate vicinity of the hole such that the presence of the tube in the stream will not affect the stream filaments in front of the impact opening. Note the shape of the tube around the impact opening in Figs. 306 and 307. Both of these tubes will give excellent service. To get a true static reading is apparently more difficult. The reason for this becomes apparent when we consider that if the static tube, which should be at right angles to the stream flow, is inclined slightly against the current, we will get too high a static reading because the true reading is increased by impact, while if the tube faces slightly down the stream, the static reading will be less than the true reading because it is affected by suction. Any irregularity in the flow of the individual water filaments in the pipe, owing to eddies, etc., will produce similar effects. Further, in some designs of Pitot tube the static tube is so placed with reference to the impact tube that the latter cannot help but disturb the stream conditions around the former, and inaccurate static readings are the result. This is no doubt why many Pitot tube measurements have proven unsatisfactory.

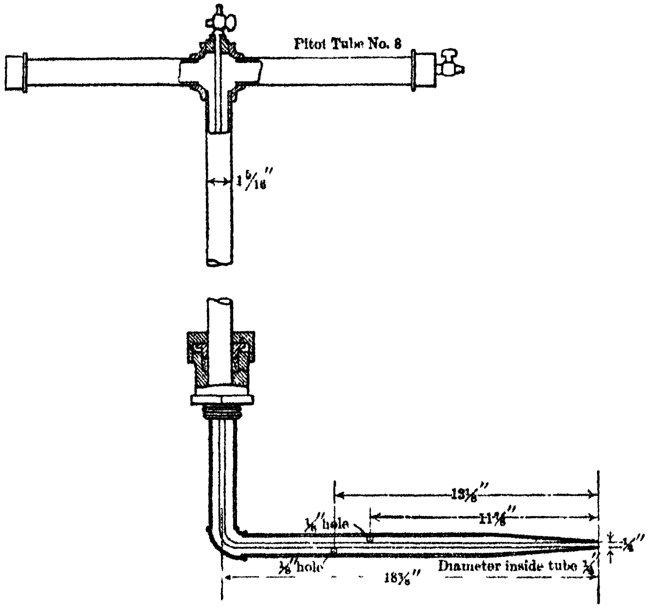


FIG. 306.—PITOT TUBE (GREGORY).

Professor Gregory, in the paper above mentioned, shows the construction of a number of tubes and discusses the results obtained

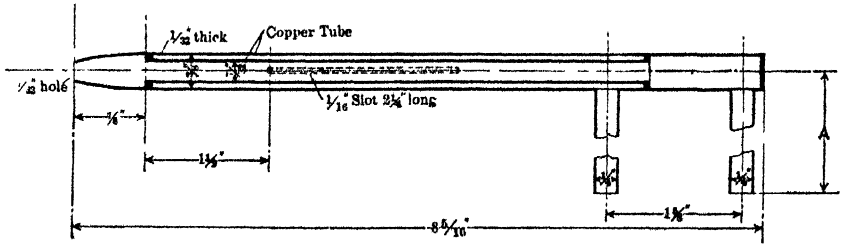


FIG. 307.—PITOT TUBE (TAYLOR).

with them. The shape of the tube finally settled upon as giving a value of $C = 1.0$, as proved by test, is shown in Fig. 306. Here, as will be seen, the inner tube is the impact tube. The static pressure is measured through small openings in the side of the outer tube. The handle at the top is used to set the tube parallel with the sides of the pipe or conduit. Another tube of similar construction is shown in Fig. 307, except that the static openings are two or three narrow slots well back from the impact opening, as shown.

Besides thus determining the proper form of tube, Professor Gregory has introduced a further simplification for tubes to be used in a straight pipe. He has shown that the static pressure, except for gravity, does not change over the cross section of a pipe or conduit. Consequently, since the static pressure can just as well be measured at the walls as anywhere else, it becomes unnecessary to have the static tube follow the impact tube when making a traverse. This means that only the impact tube need be introduced, and if the tube is of the right shape the only openings necessary in any conduit will be, say, a 1½ inch opening for the impact tube and two ⅛ inch or ¼ inch openings 90 degrees from the former and located at the ends of a diameter for the static reading. Into the latter are screwed ¼ inch valves, but care should be taken first to round off the inner edges of these holes with a rat-tail file and the valves should *not* be screwed through the thickness of the wall. This prevents suction or impact at these openings. The Pitot tube itself is pushed through a packing gland in a plug which is provided with ordinary pipe thread. The tube is provided with a handle, by means of which it is raised or lowered and also aligned with the pipe. The connections are then as shown in Fig. 308, Fig. 309 showing the lower end of the tube on a larger scale. The form of differential manometer shown is also very convenient. At *A* there may be connected a force pump by means of which the position of the water levels in the manometer may be controlled.

215. Calibration of Pitot Tubes. — The only methods of calibrating Pitot tubes are the same as those available for current meters; that is, they must either be dragged through still water at a definite rate or they may be placed in streams the velocity of which is known. The experimental work done, however, has shown that where the best form of tube is used with care, the constant may be assumed equal to 1.0. The error involved in this assumption is almost sure to be not greater than ± 1 per cent.

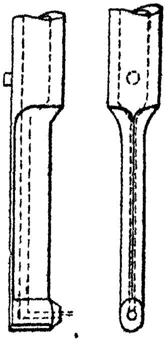
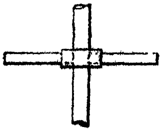


Fig. 309.



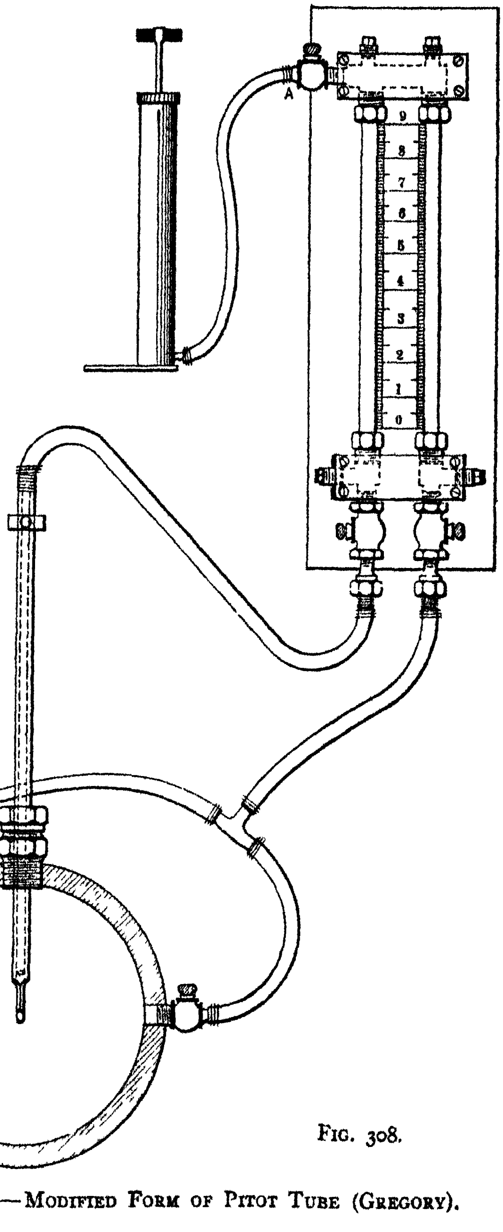


Fig. 308.

Figs. 308 and 309.—Modified Form of Pitot Tube (Gregory).

216. Method of Making a Traverse and Obtaining the Average or Mean Velocity. — It has been shown by a number of tests that the velocity of water going through a pipe is greatest at the center and least at the walls, and that the ratio of these two velocities approximates two to one. The variation of the velocities may be represented by a solid of revolution whose axial cross section consists of a rectangular part *ABDE*, Fig. 310, with a semi-elliptical end *BCD*. The rectangle has for its length one-half the center velocity *FC′*, while the other dimension is the diameter of the pipe *AE*. The semi-ellipse has for its minor axis the diameter of the pipe and for half its major axis *CC′*, one-half of the center of velocity *FC′*. How closely the ellipse is approximated is well shown in Fig. 310.

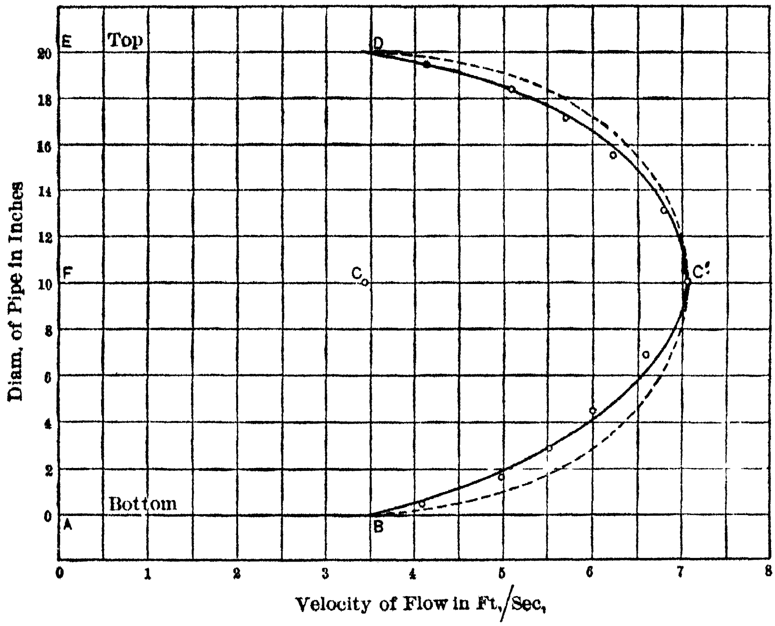


FIG. 310. — DISTRIBUTION OF VELOCITIES IN A PIPE.

For the purpose of determining the complete shape of the cross section *ABC′DE*, the pipe must be traversed. This may be done by taking readings at stations equally spaced along a diameter, in which case the curve must be plotted and the mean velocity determined by finding by integration the mean ordinate of the solid of revolution. Or certain definite stations may be chosen so that the velocity readings may be averaged straight without going through the process of drawing the velocity curve. The last method is the quicker and fully as accurate as the former. Any number of stations may be chosen along a given diameter, but it has been shown that ten stations give an average sufficiently close. Hence this method of determining average velocity is sometimes known as the 10-point method. In Fig. 311 the cross section of the pipe is divided into five *equal* concentric areas indicated by the dotted circles. Each area is then divided into two equal areas indicated by the short circular arcs, resulting in "Stations" 1 to 10 (marked by heavier dots) along the diameter *A B*. The reading at each one of the stations 1–5 will give the mean velocity for the concentric area in which it is located. Stations 6–10 simply duplicate the readings taken in the upper half of the pipe, so that if a complete traverse is made, two velocity head readings will be taken in each area. Since the areas are equal, the sum of all the velocities for any one traverse is simply divided by 10 to obtain the average velocity. Note that the center velocity, while it should be read on the traverse, does not enter this computation. Fig. 311 shows how the stations may be determined. In practice a scale may be laid off on the Pitot tube itself so that the adjustments may be made quickly.

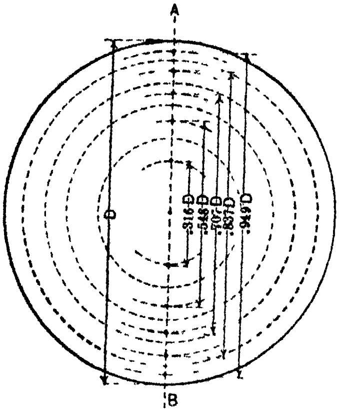


FIG. 311.

217. Relation between Center Velocity and Mean Velocity. — Assuming that the velocity traverse curve is a true ellipse, the mean velocity would be .833 × the center velocity. Gregory and others have in actual cases found this factor to be about .84. In any case this ratio gives a very quick means of approximately determining flow from a single Pitot tube reading of the center velocity, but it should be noted that the error may be ± 3 per cent.

218. The Flow of Water in Pipes. — Many of the factors governing the flow of water in pipes are as yet not definitely determined, and the computations are therefore to a certain extent approximate. For a full discussion of this matter see any book on Hydraulics or on Mechanics containing chapters on Hydromechanics.

In general, if h is the total head in feet available to produce flow, and V is the velocity of the issuing stream from the pipe (not nozzle) in feet per second, we may write

$$h - \frac{V^2}{2g} = h_1 + h_2 + h_3 + h_4, \tag{27}$$

in which the right-hand member of the equation expresses the sum of all the losses of head. Here

h_1 = the loss of head at entrance.
h_2 = the loss due to skin friction in the pipe.
h_3 = the loss of head due to curves and bends.
h_4 = the loss of head due to other causes, such as obstructions, valves, etc.

Each one of the losses may be expressed by a term equal to $\frac{V^2}{2g}$ multiplied by some correction factor ϕ, except the loss due to skin friction, a general expression for which is $h_2 = 4f\frac{l}{d}\frac{V^2}{2g}$, where l is the length and d the diameter of the pipe in feet, and f may be considered as a coefficient of friction.

Hence equation (27) may be written:

$$h - \frac{V^2}{2g} = \phi_1\frac{V^2}{2g} + 4f\frac{l}{d}\frac{V^2}{2g} + \phi_3\frac{V^2}{2g} + \phi_4\frac{V^2}{2g}, \tag{28}$$

where V is in every case the velocity in feet per second at that place in the pipe for which the loss is being computed. Where the pipe is made up of lengths of different diameters, or where bends of various curvatures are used, or where several kinds of valves are employed, each one of the factors h_2, h_3 and h_4 stands for the sum of all the individual losses coming under that particular classification.

Value of ϕ_1. — This may be taken equal to .5 for a pipe at right angles to the reservoir or source of supply and flush with the inside wall, and where the entrance edges are sharp. These are about the conditions that exist when a pipe is screwed into a tee in the main pipe. Where the pipe projects inside, ϕ_1 may be as high as .93, while if the entrance is bell-mouth, ϕ_1 may be almost = 0.

Value of f. — For values of the coefficient of friction *f*, see the curves in Fig. 312.[*] These figures are for clean pipe. Note that they vary with the size of the pipe and the velocity *V*. For rough or foul pipe the value of *f* should be increased. The amount of increase varies with the kind of pipe and is not definitely known. For a slightly rough pipe an increase of 30 per cent, while for a very

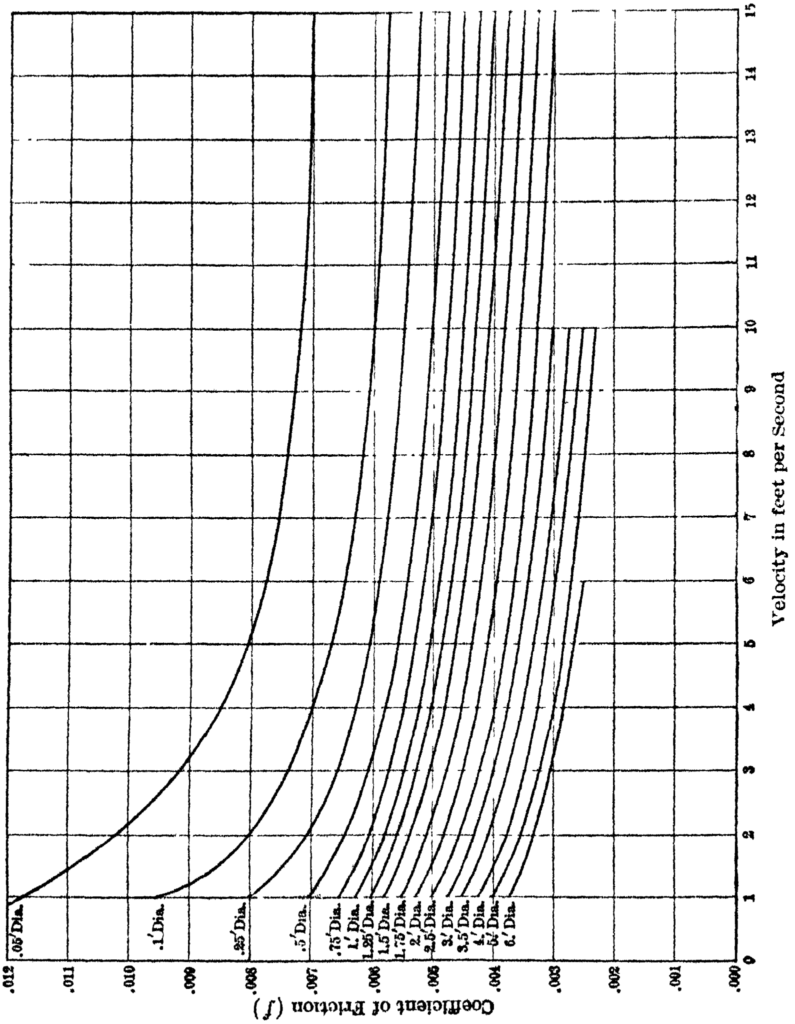


FIG. 312. — COEFFICIENTS OF FRICTION FOR FLOW OF WATER IN PIPES.

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[*] Adapted from a table given in Merriman's "Hydraulics."

foul pipe one of 100 per cent may serve the purpose, although the matter must be left to the judgment of the engineer.

Value of ϕ_3. —For this factor various authorities also give information somewhat at variance. Weisbach found

$$\phi_3 = .9457 \sin^2\frac{\theta}{2} + 2.047 \sin^4\frac{\theta}{2},$$

where θ is the exterior angle (ABC, Fig. 313). From this, when

$\theta =$	$22\frac{1}{2}°$	$45°$	$90°$
$\phi_3 =$	.038	.181	.984

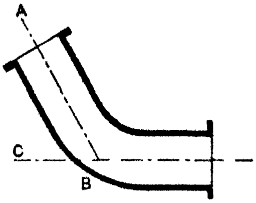

FIG. 313

These are experimental figures found for $1\frac{1}{4}$ inch pipe. For smaller pipe the values increase. Thus Weisbach found for a $\frac{3}{8}$ inch pipe a coefficient of 1.53 for a 90-degree elbow. On the other hand, for larger pipe the values for ϕ_3 may be expected to be less than those given.

In long bends, instead of elbows, the friction loss is much less, and the greater the ratio $\frac{R}{r} = \frac{\text{Ratio of curvature}}{\text{radius of pipe}}$, the smaller the loss. Thus for 90-degree bends Weisbach found, if

$\frac{R}{r} =$	10	5	$2\frac{1}{2}$	$1\frac{1}{4}$	1
$\phi_3 =$	0.131	.138	.206	.977	1.978

Some authorities content themselves by simply adding for each elbow in a pipe line a certain number of extra feet to the length l of the line to allow for the extra friction. Thus Latta gives the following allowances:

Nominal Diam. of Pipe, inches,	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5
Allowance in ft. for each elbow,	2	3	5	7	9	11	13	19

Such allowances are of course empirical, since they do not take into account the velocity of the water.

Value of ϕ_4. —Perhaps least is known about the value of this coefficient. Weisbach gives figures for gate valves, plug cocks, and butterfly valves, but not for the globe valve so much used in this country. For a gate valve the figures are as follows (see Fig. 314):

Ratio $\frac{d'}{d}$ =	0	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
ϕ_4 =	0	.07	.26	.81	2.1	5.5	17	98

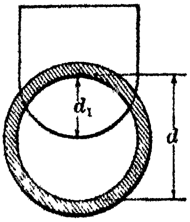

FIG. 314.

No definite figures for globe valves appear to be available. Some writers assume the resistance of such a valve, when open, at about 1.5 times that of a 90-degree elbow.

In practice the case encountered in the great majority of instances is that of a pipe of uniform diameter. For this case the velocity V in all the terms of equation (28) may be taken the same and the equations may be solved for V, the mean velocity in the pipe. We will then have

$$V = \sqrt{\frac{2\,gh}{1 + \phi_1 + 4f\frac{l}{d} + \phi_3 + \phi_4}} \text{ ft. per sec.} \tag{29}$$

The relative importance of the terms varies with the length l of the pipe, as the following example will show. In many cases all but $f\frac{l}{d}$ and perhaps ϕ_3 may be neglected.

Example 1. Compute the discharge through a 3-inch wrought-iron pipe slightly incrusted. The pipe is 1000 feet long in all, has 6 right-angled turns (ordinary elbows) with a globe valve at the end. The pressure of water at the main is 30 pounds by gauge and the discharge end of the pipe is 10 feet below the point of connection to the main.

Here the total head $h = 2.3 \times 30 + 10 = 79$ feet. The value of ϕ_1 may be taken equal to .5. Since equation (29) involves the inter-related factors V and f, it must be solved by trial, and for a first approximation put $f = .006$, for which $V = 6$ feet per second (see Fig. 312). The length $l = 1000$ feet, and $d = 3$ inches $= .25$ feet. ϕ_3 may be taken equal to .984, and since there are six elbows, the total loss will be $6 \times .984 = 5.90$. For the globe valve put $\phi_4 = 1.5 \times .984 = 1.5$.

Then

$$V = \sqrt{\frac{2 \times 32.2 \times 79}{1 + .5 + .006 \times 4\,\frac{1000}{.25} + 5.9 + 1.5}}$$

$$= \sqrt{\frac{5087}{1 + .5 + 96 + 5.9 + 1.5}} = \sqrt{\frac{5087}{104\,9}} = 6.9 \text{ feet per second.}$$

This is in close enough agreement with our assumption for all practical purposes. Since the internal area of a 3-inch pipe is .0513 square feet, the cubic feet discharged will be $6.9 \times .0513 = .35$ per second.

An analysis of the above example will show that approximately 90 per cent of the loss of head is accounted for in skin friction and about 5.5 per cent is accounted for in the loss through the six elbows. Neglecting all losses except that due to skin friction, the result would have been $V = 7.2$ feet per second, which is within 5 per cent of the former result. Hence for very long pipe it is only necessary to take skin friction into account and all other losses may be neglected, provided the number of curves or bends is not excessive.

Case of Short Pipes. — A pipe is regarded as short when its length is less than 500 times the diameter and very short when it is less than 50 diameters. In such a case with high heads, V is apt to be so great that little is known about the value of f. It also becomes necessary to determine ϕ_1 as closely as possible. For the conditions existing and where the pipe is very short, a length equal to three diameters should be subtracted from the length l, since that part of the length is accounted for in the value of ϕ_1.

Example 2. Conditions same as in Example 1 except that the pipe is only 50 feet long. The statement then becomes, assuming for the moment that f remains the same,

$$V = \sqrt{\frac{2 \times 32.2 \times 79}{1 + .5 + .006 \times 4 \frac{50}{.25} + 5.9 + 1.5}}$$

$$= \sqrt{\frac{5087}{1 + .5 + 4.8 + 5.9 + 1.5}} = 19.3 \text{ ft. per sec.}$$

Now the curves, Fig. 312, for this case indicate that the value of f for this velocity is probably much nearer .005. With this value we get

$$V = \sqrt{\frac{5087}{1 + .5 + 4 + 5.9 + 1.5}} = 19.9 \text{ ft. per sec.}$$

Note that in this case the head lost in friction is smaller than the head lost in the elbows and that none of the losses can be neglected.

Measurement of Gases.

219. As in the case of liquids, the methods of measuring gas may be positive or inferential. In all cases positive methods should be used where possible, on account of greater guarantee of accuracy.

The various possible methods available may be grouped as follows:

I. Chemical Methods. Used positively for the measurement of small quantities of gas. Used also inferentially in boiler and producer testing. In either of the last two applications only a small part of the flue or producer gas is analyzed, and from this with the aid of other factors the total quantity of gas concerned in a given operation may be determined. The reader is referred to Chapters XIII and XXI for discussion of these analyses.

II. Measurement by means of orifices, such as openings in flat plates, nozzles, etc. These methods are equally applicable to large and small volumes and among them are some of the best available.

III. Measurement by volume displacement, as in gasometers or gas meters in general.

IV. Measurement by determining the average velocity of flow in a pipe of known cross section by means of anemometers, Pitot tubes, Venturi meters, etc.

V. Measurement by calorimetry, or by steam and electric heaters.

In measuring gases there are several things which must be constantly borne in mind. They are:

1. Volume means nothing unless temperature and pressure at which the volume is measured are determined and recorded.

2. Pressure measurement means nothing unless the degree of saturation of the gas with water vapor is known. All gases as handled by the engineer are more or less nearly saturated with water vapor and the pressure exerted by such a mixture is the sum of that due to the gas and that due to the water vapor. In making refined calculations the pressure due to the water vapor must be subtracted from that of the mixture giving the "partial pressure" due to the gas; and the pressure of water vapor in question is *not* that given in the steam tables as corresponding to the temperature, but some fraction of that, the fraction being the "percentage humidity." This corrected partial pressure is then used in the ordinary gas formulas.

3. Methods of measurement in which the gas to be measured comes in contact with water or similar liquid are always subject to error, due to the solubility of the different gases in the liquid. Any liquid to be used in this way should be saturated with the gas or gases in question by allowing them to bubble through it for a considerable length of time before measurements are begun.

Where the quantity of liquid is relatively small, as in a wet gas meter, it soon becomes saturated with the gas and its effect may thereafter be disregarded unless there are considerable changes of temperature.

The following table gives the cubic feet of gas measured at 60° F. and 14.7 pounds per square inch which are soluble in one cubic foot of water at 14.7 pounds pressure and different temperatures.

Temp., Deg. F	32°	68°	212°		32°	68°	212°
Air........	0.027	0.0185	......	Carbon monoxide..	0.037	0.025	....
Nitrogen.......	0.026	0.017	0.0105	Carbon dioxide. ..	1.06	0.98	0.26
Oxygen......	0.053	0.034	0.0185	Ammonia.........	1250	700	......
Hydrogen.. . .	0.023	.020	0.018				

II. THE MEASUREMENT OF GAS BY MEANS OF ORIFICES.

220. Theoretical Velocity and Weight Discharged. — The flow is assumed adiabatic. Where the work must be so close that friction effects must be allowed for, this is generally done by applying experimentally determined coefficients to the results obtained by adiabatic computations.

Following the methods of Peabody,* suppose the vessel A, Fig. 315, contains a frictionless piston B and the connecting tube C a frictionless piston D. Let the absolute pressure on B be P_1 pounds and the pressure on D be P_2 pounds per square foot. As B moves to the right, each pound of gas has done upon it the work P_1v_1, while each pound of gas entering C does the work P_2v_2, v_1 and v_2 being the specific volumes in cubic feet of the gas for the respective absolute pressures P_1 or P_2 per square foot. If the velocity of the gas in $A = V_1$, and the velocity in the orifice $C = V_2$, the kinetic energies possessed by one pound of the fluid will be $\frac{V_1^2}{2g}$ and $\frac{V_2^2}{2g}$ respectively. Let the intrinsic energy in A be

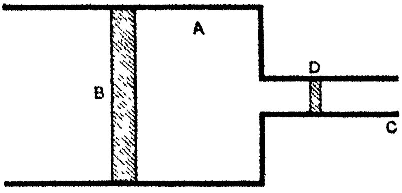

FIG. 315.

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* Thermodynamics of the Steam Engine, p. 424.

E_1 and that in C be E_2. Then it follows, from the assumption of adiabatic flow, that

$$E_1 + P_1v_1 + \frac{V_1^2}{2g} = E_2 + P_2v_2 + \frac{V_2^2}{2g}. \tag{30}$$

With well-insulated nozzles or pipes the condition of adiabatic flow is very nearly realized, even if the pipe is of considerable length.

In practice it is customary, where orifices are used for measurement, to so proportion the size of A to the size of the orifice that V_1 shall be very small as compared with V_2. Hence the terms depending upon the former can usually be neglected, and solving for V_2 we may write,

$$V_2 = \sqrt{2g(E_1 - E_2 + P_1v_1 - P_2v_2)} \text{ ft. per sec.} \tag{31}$$

Now it can be shown that all the intrinsic energy in a gas is available for outside work in an adiabatic change and that in general

$$E = \frac{Pv}{\gamma - 1}.$$

Making the substitution in (31) we then have

$$V_2 = \sqrt{2g\left(\frac{P_1v_1}{\gamma - 1} - \frac{P_2v_2}{\gamma - 1} + P_1v_1 - P_2v_2\right)}$$

$$= \sqrt{2g\left[P_1v_1\left(\frac{1}{\gamma - 1} + 1\right) - P_2v_2\left(\frac{1}{\gamma - 1} + 1\right)\right]} \text{ ft. per sec.} \tag{32}$$

From the adiabatic law we may derive

$$P_2v_2 = P_1v_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

and substituting this value of P_2v_2 in (32) we obtain

$$V_2 = \sqrt{2gP_1v_1\frac{\gamma}{\gamma - 1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \text{ ft. per sec.} \tag{33}$$

Since temperatures are quite readily determined, it is more convenient *to substitute for* P_1v_1 *in the above equation the equivalent* RT_1, *where* R *is the gas constant and* T_1 *the absolute temperature* in the reservoir. Hence we will finally have the theoretical velocity

$$V_2 = \sqrt{2gRT_1\frac{\gamma}{\gamma - 1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \text{ ft. per sec.} \tag{34}$$

With the aid of formula (34) it becomes easy to find the equations for theoretical discharge. If F is the area of the orifice in square feet, the theoretical discharge in cubic feet will be

$$Q = FV_2 = F\sqrt{2gRT_1\frac{\gamma}{\gamma-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}\text{ cu. ft. per sec.,} \tag{35}$$

and further, if δ is the density of the gas, in pounds per cubic foot, ***under the condition for which V_2 is determined,*** the weight theoretically discharged will be

$$W = F\delta\sqrt{2gRT_1\frac{\gamma}{\gamma-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]}\text{ lbs. per sec.} \tag{36}$$

In practice, equation (36) must be multiplied by a coefficient of discharge C, which depends upon the kind of orifice and for which values will be given later.

221. Value of the Pressure P_2. — So far nothing has been said of the value of the pressure to which the gases expand in the nozzle or orifice. It has been tacitly assumed that P_2 is always the pressure in the discharge space beyond the orifice. Experiment, however, has shown that the flow under some conditions follows laws which do not correspond with this assumption. An examination of equation (36) will bring out the error of such a supposition also on theoretical grounds. Assume in one case that the pressure in the discharge space is equal to P_1. Then equation (36) will give $W = 0$, and that is a logical result, for there is then no pressure difference. Next assume that the pressure in the outside space is 0, that is, that there is a perfect vacuum. But if P_2 could drop to that value, equation (36) would again give $W = 0$, because δ would then be equal to 0. This conclusion is absurd, because under these conditions we should rather expect maximum discharge. Hence P_2, the pressure in the orifice, does not in all cases fall to the pressure in the outside space.

It can be shown that there is an increase in the flow as P_2 decreases below P_1 until a certain maximum discharge is reached.

It can also be shown mathematically that this condition is obtained when

$$\frac{P_2}{P_1} = \left(\frac{2}{\gamma - 1}\right)^{\frac{\gamma}{\gamma-1}}. \qquad (37)$$

The value of P_2 obtained from this expression, for a given P_1, is called the *critical pressure*, and the ratio $\frac{P_2}{P_1}$ the *critical ratio*. Its theoretical value for air and some other gases, for which γ is near 1.4, is $\frac{P_2}{P_1} = .527$.

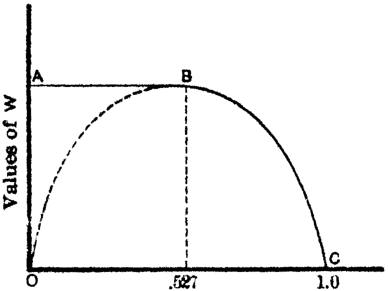

FIG. 316.

Should the pressure in the discharge space drop below the critical value of P_2, the latter shows no further decrease but retains its value .527 P_1. Graphically this condition of things may be represented as in Fig. 316, in which ABC represents the curve of discharge. From C to B, as P_2 decreases, the rate of discharge increases until $\frac{P_2}{P_1} = .527$. Theoretically, W should then again decrease to 0 when $\frac{P_2}{P_1} = 0$, as shown by the branch BO. Instead of this, P_2 becomes independent of the outside pressure and W becomes sensibly constant, as shown by the line BA.

In any practical case the method of computation is then as follows:

(*a*) Determine the ratio of $\frac{\text{Pressure in discharge space}}{\text{Pressure in reservoir}}$.

(*b*) If this is equal to or greater than .527, substitute in equations (34), (35) and (36) for P_2 the value of the pressure in the discharge space.

(*c*) If the ratio is less than .527, substitute in the equations for P_2 a value equal to .527 of the reservoir pressure.

222. Formulas for the Flow of Air. — Fixing upon the kind of gas, air in this case, allows of the substitution in equations (34), (35) and (36) for the constants $g = 32.2$, $R = 53.3$ and $\gamma = 1.405$.* The equations then become

$$V_2 = 109.1\, C_v \sqrt{T_1\left[1 - \left(\frac{P_2}{P_1}\right)^{.29}\right]} \text{ ft. per sec.} \qquad (38)$$

$$Q = 109.1\, CF \sqrt{T_1\left[1 - \left(\frac{P_2}{P_1}\right)^{.29}\right]} \text{ cu. ft. per sec.} \qquad (39)$$

and

$$W = 109.1\, CF\delta \sqrt{T_1\left[1 - \left(\frac{P_2}{P_1}\right)^{.29}\right]} \text{ lbs. per sec.} \qquad (40)$$

in which C_v and C are the coefficients of velocity and of discharge respectively. Within the assumptions made, these formulas are exact and must be used whenever there is any considerable change of density in passing from P_1 to P_2. When the change of density is small, that is, when the ratio $\frac{P_2}{P_1}$ is, say, not less than .9, there are certain substitutions that may be made which simplify the numerical work involved without introducing great errors. But judgment must be exercised in the use of such approximate equations. In this connection see papers by S. A. Moss in *Power* for Sept. 20 and 27, 1906, and by R. J. Durley, Trans. A.S.M.E., 1905.

Note particularly that P_2 is the pressure in the discharge space only as long as the latter is not less than .527 P_1. For the condition of lower pressures than this in the discharge space, P_2 remains at the critical value, and the area F is that cross section of orifice or nozzle at which P_2 is established. Further, δ is the density according to the pressure P_2 and the temperature T_2. Unless P_2 is also the pressure in the discharge space, T_2 is hard to determine, and should be substituted for in terms of P_1, T_1 and R. This can be done as follows: From a combination of the adiabatic law and of the equation of state for gas, we may write

$$\delta = \frac{1}{v_2} = \frac{1}{v_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} = \frac{P_1}{RT_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}}$$

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* γ change with temperature, see Chapter XXI under "Specific Heat."

Substituting this in the general theoretical equation (36) and in the special equation (40), we obtain

$$W = FP_1 \sqrt{\frac{2g}{RT_1}\frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right]} \text{ lbs. per sec.} \qquad (41)$$

and $$W = 2.047\ CFP_1 \sqrt{\frac{1}{T_1}\left[\left(\frac{P_2}{P_1}\right)^{1.42} - \left(\frac{P_2}{P_1}\right)^{1.71}\right]} \text{ lbs. per sec.} \qquad (42)$$

Equations (41) and (42) are probably the most serviceable, the former giving the theoretical discharge for any gas, the latter the actual discharge for air and other gases for which γ = about 1.405. The following table gives values of γ for some of the more common gases, for ordinary conditions of pressure and temperature.

For air, $\gamma = 1.402$
oxygen, $\gamma = 1.400$
atmospheric nitrogen, $\gamma = 1.408$
hydrogen; $\gamma = 1.420$
CO, $\gamma = 1.408$
CO_2, $\gamma = 1.296$

223. Value of the Coefficient of Discharge C. — No definite information seems to be available concerning the value of C for high-pressure work. For low pressures the following figures may be used. For high-pressure work, and indeed for all accurate work, each orifice should be specially calibrated. The scheme mentioned in Art. 224 may sometimes be used when it is desired to avoid calibration in connection with high pressures.

For the purpose of measurement it seems best, judging from the data now available, to choose a bell-mouth orifice. This becomes apparent upon examination of the values given below for the constant C. A well-rounded orifice has a constant close to .95 and, above all, this figure does not change materially with a variation in the ratio $\frac{P_2}{P_1}$. Both Hirn and Rateau have shown this, the former for air and the latter for steam. For an orifice in a thin plate or for short cylindrical mouth-pieces with sharp edges, the discharge constant varies with the pressure ratio. It may be near .6 for pressure ratios in the neighborhood of .9 or above and may rise to .8 for pressure ratios near .5. The work of Hirn and Rateau confirm this, and Weisbach also found this, as indicated below.

Weisbach determined the following values of C for air:

Conoidal mouth-piece of the form of the contracted vein, with effective pressures of 0.23 to 1.1 atmospheres	0.97 to 0.99
Circular sharp-edged orifices	0.563 to 0.788
Short cylindrical mouth-pieces	0.81 to 0.84
The same rounded at the inner end	0.92 to 0.93
Conical converging mouth-pieces	0.90 to 0.99

Moss* found the coefficient of discharge C to be on the average .942 for moderate pressures, say up to 8 pounds above the atmosphere, and for a well-rounded orifice 1 inch in diameter, the straight portion having a length of about 1½ inches.

Prof. R. J. Durley, in the paper above mentioned, gives coefficients of discharge for circular orifices in thin plates up to 4.5 inches diameter for pressures up to 5 inches water. (See table.) The thickness of the orifice plates was .057 inch. The orifices were placed in the end wall of a long box of considerably greater cross section, so that the initial velocity could be neglected. For consistent results it was found that the cross section of the box should be at least 20 times that of the orifice. Pressure and temperature were measured about 5 feet back from the orifice, the manometer connection

Diameter of Orifice, Inches.	Pressure in Inches Water.				
	1	2	3	4	5
$\frac{5}{16}$ . . . $C =$	603	.606	610	.613	.616
$\frac{1}{2}$	602	.605	.608	.610	.613
1	.601	.603	.605	.606	.607
1½	601	.601	602	.603	.603
2	.600	.600	.600	.600	.600
2½	.599	.590	.599	.598	.598
3	.599	.598	.597	.596	.596
3½	.599	.597	.596	.595	.594
4	.598	.597	.595	.594	.593
4½	598	.596	.594	.593	.592

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* S. A. Moss, *Power*, Aug. 10, 1905.

being carried through the wall so as to be flush with the inner surface.

The older work of Hirn* on the flow of air through orifices and nozzles showed that for a properly designed nozzle the discharge coefficient C varied but slightly with pressure ratios varying from $\frac{P_2}{P_1} = .52$ to $.99$, and found $C = .966$ to $.985$ for a nozzle with 9 degrees convergence, and $C = .980$ to $.991$ for a nozzle with 13 degrees convergence.

For an orifice in a thin plate he found the following results:

$\frac{P_2}{P_1} =$	.95	.9	.8	.7	.6	.5	.4
$C =$	.627	.640	.674	.708	.741	.773	.799

This series is recomputed from the values given by Hirn, and represents the ratio of the discharge volume of the thin-plate orifice to the theoretical discharge volume of a bell-mouth orifice (not to the discharge volume of a convergent nozzle, as Hirn gives it).

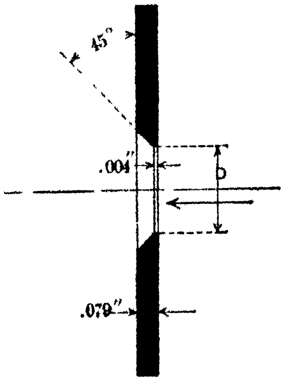

FIG. 317.

One of the most recent investigations on this subject is that made by A. O. Müller and reported in the *Zeitschrift des Vereins deutscher Ingenieure*, Feb. 22, 1908. The paper describes the testing apparatus and the equations applying to each case in detail. The pressure differences being small, Müller neglected differences of density in the gas on the two sides of the orifices and used the simplified formulas $V = CF\sqrt{2gh}$ or $V = KF\sqrt{2gh}$ (see below), in which V is volume in cubic feet per second and h is the pressure difference expressed in feet of gas column at the density corresponding to the reservoir pressure. It should therefore be remembered that the coefficients quoted below apply to low-pressure work only.

The dimensions of the orifice used are given in Fig. 317. Four cases were investigated:

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* *Annales de Chimie et de Physique*, March, 1886.

(1) Flow from the receiver through the orifice direct into the atmosphere.

(2) Flow from the receiver through the orifice into a tube having an internal diameter of 3.22 inches.

(3) Flow through an orifice, placed at the end of a tube having an internal diameter of 3.22 inches, into the atmosphere.

(4) Flow through an orifice placed across a pipe having an internal diameter of 3.22 inches.

The conditions of flow are indicated in Figs. 318 to 321. In cases (2), (3) and (4) the flow is affected by the varying ratios of the area of orifice F to the tube area F_1. This ratio $\frac{F}{F_1}$ in the following tables is indicated by m.

The pressure differences used in this work did not exceed about 4 inches water. The results obtained by Muller for case (1) agree very well with those found by Durley. In the other three cases the constant is largely affected by the conditions, that is, by the ratio m and the friction in the length of the tube l, the latter being measured as shown. The constant is in the latter cases radically different from the constant C applying to the thin-plate orifice alone, and has been designated therefore by the letter K. There is a certain relation between C and K by means of which in any given case, after C is determined from experimental data available, K may be computed, taking into account the ratio m and the friction loss in the length l. For the derivation of this relation see the paper mentioned. Below the equation is given for each case.

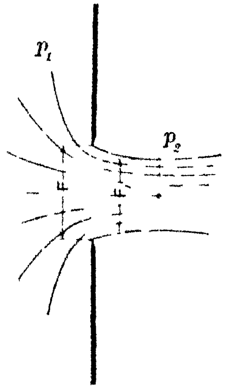

FIG. 318.

Case (1)

Diameter of orifice, inches,	.92	1.73	2.44
Range of pressures above atm., in. H_2O,	.37–2.32	.29–1.45	.21–.96
Mean coefficient C,	.597	.598	.596

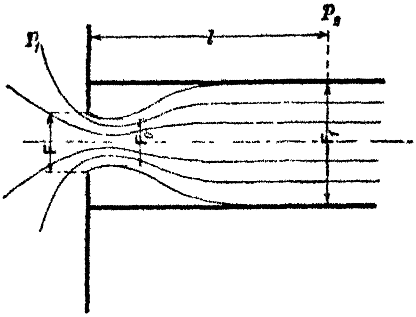

FIG. 319.

Case (2)

$$\frac{1}{K} = \sqrt{(1 + fl)\, m^2 + \left(\frac{1}{C} - m\right)^2}$$

where l is expressed in feet, f may be taken equal to .124.

Diameter of orifice, inches,	.92	1.73	2.44
Range of pressures above atm., in. H_2O,	.19–2.2	.21–2.2	.04–2.8
$\frac{F}{F_1} = m =$	.082	.287	.578
K (determined),	.632	.685	.764
C (computed),	.602	.585	.583

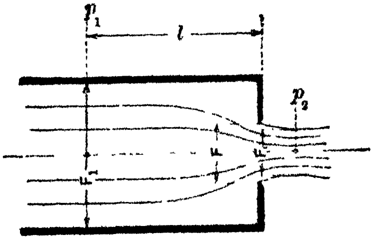

FIG. 320.

Case (3)

$$K = \frac{C}{\sqrt{1 - (1 - fl)\, m^2C^2}}$$

l and f same as in case (2).

Diameter of orifice, inches,	.92	1.73	2.44
Range of pressures above atm., in. H_2O,	.12–4.2	.16–3.8	.12–4.0
$\frac{F}{F_1} = m =$	.082	.287	.578
K (determined),	.603	.644	.755
C (computed),	.602	.633	.691

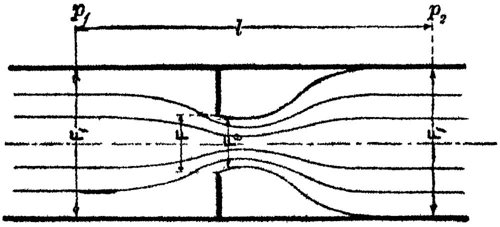

FIG. 321.

Case (4)

$$\frac{1}{K} = \sqrt{flm^2 + \left(\frac{1}{C} - m\right)^2}$$

l and f same as in case (2).

Diameter of orifice, inches,	92	1.73	2.44
Range of pressures above atm., in. H_2O,	.20–3.4	.12–3.5	.07–2.5
$\frac{F}{F_1} = m =$	.082	.287	.578
K (determined),	.641	.750	1.084
C (computed),	.609	.621	.697

224. Necessary Precautions in Orifice Methods. — Great care must be taken in measuring air by means of orifices that flow may take place and observations may be made in the same way as in the original experiment for the determination of the constant. It is best, when possible, to consult the original report in order to be able to correctly reproduce the conditions.

In general, attention must be given to the following points:

(1) The orifices must be carefully and accurately made and correctly measured.

(2) The velocity of approach must be reduced to a negligible value.

(3) The pressure must be measured in such a way as to get least disturbance of flowing gas. The common method is to connect the manometer to a tube which passes perpendicularly through the walls of the reservoir or pipe just ahead of the orifice. The inner end of the tube must be flush with the inner surface of the wall.

(4) Leakage must be prevented or its value determined and corrections made.

(5) All changes of section immediately preceding the orifice should

be avoided when possible, and if absolutely necessary should be gradual and rounded.

(6) Steady conditions of flow must, if possible, be secured during measurements.

With proper precautions this method is capable of giving results accurate to 1 or 2 per cent, and with extreme care an accuracy well within 1 per cent can be obtained.

For measurement by means of orifices when the pressure differences are greater than those for which reliable values of the constants are available, it is only necessary to reduce the high reservoir pressure by throttling into another vessel from which the flow to the orifice takes place. Care should then be taken to insure against disturbances caused by throttling the discharge. This can always be done by allowing sufficient distance between the orifice and the point at which throttling takes place.

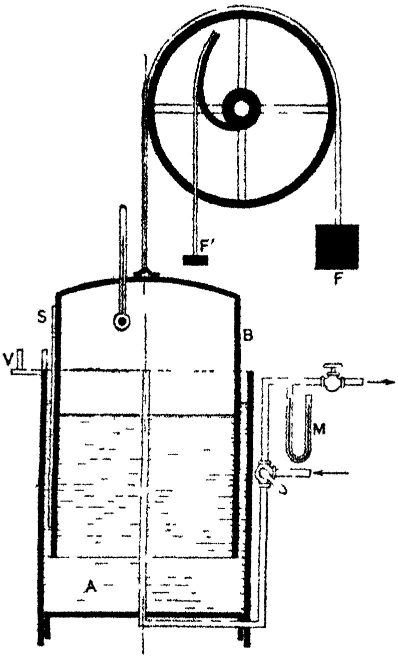

FIG. 322.—GASOMETER

III. MEASUREMENT BY MEANS OF VOLUME DISPLACEMENT, GAS METERS, ETC.

225. The Gasometer.—The simplest form of gas meter is the well-known gas holder, or so-called *gasometer*. This apparatus consists essentially of a tank *A*, Fig. 322, containing water or other liquid into which dips an inverted tank *B* called the bell. The piping may be arranged as in Fig. 322, or separate inlet and outlet pipes may be used. The bell is either entirely or partly counterbalanced by a weight *F*, depending upon the gas pressure desired. This method of counterbalancing, if used alone, would result in a pressure change in the gas as the bell ascends or descends on account of different degrees of immersion of the bell. To allow for this a compensating weight *F'* acting on a changing arm is used where accurate work is essential. To definitely determine weight or volume of gas with this instrument, three observations are necessary: Pressure, temperature and movement of bell in a given time. Any suitable instrument may be employed for the former, although since the pressure is usually but a few inches above atmosphere the ordinary water manometer is mostly used, see *M*, Fig. 322. To determine temperature accurately it is essential that the temperature of the gasometer liquid should not be far different from that of the gas. The movement of the bell can usually be read directly by means of a suitable scale *S* fastened to the bell, while the tank supports a pointer or sighting arrangement *V*. This measurement is susceptible of great refinement by electrical means, if that should be required. Where the bell is not

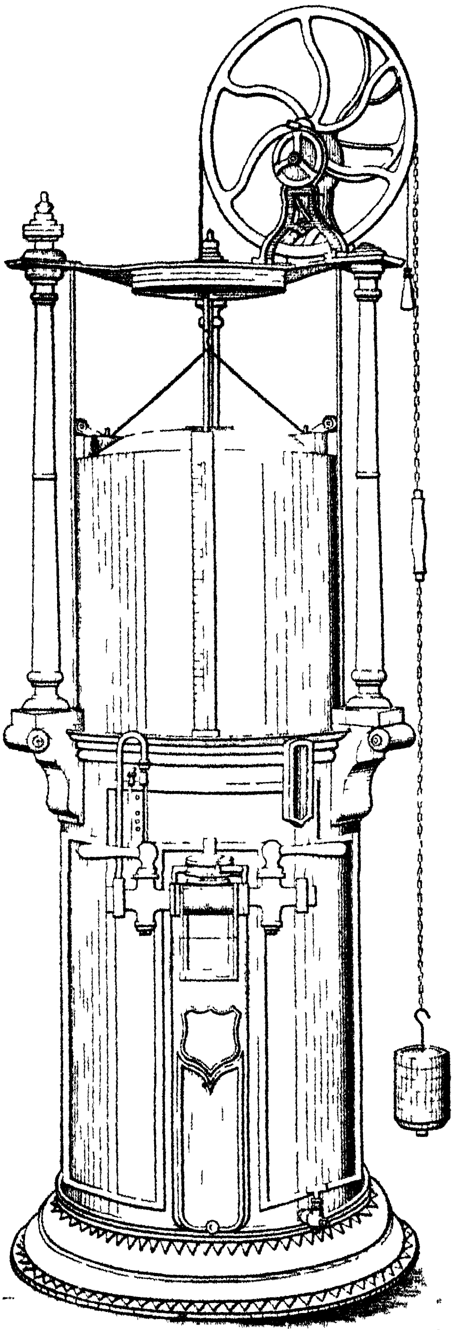

FIG. 323 —METER PROVER.

large and accurately guided, a reading of the movement of the bell at a single station may be sufficient, otherwise simultaneous readings at not less than three stations are recommended.

Gasometers may be calibrated either by computation or by actual experiment. The former method should be used only where the bell is comparatively small, so that the diameter can be accurately measured, and where it is accurately cylindrical. Where the second method of calibration can be applied it should always be used in preference to the first.

Gasometers are used in two ways, — to measure gas from a producer or to a consumer, and largely also to calibrate other types of meters, in which case they are usually called *meter provers.*

The former method of service needs no further explanation. It is intermittent unless two gasometers are used in parallel, one emptying while the other is filling.

Meter provers are simply accurately constructed and carefully calibrated gasometers. A type of one of these in elevation is shown in Fig. 323.

226. Gas Meters. — It was stated above that a single gasometer cannot be used for continuous measurement. Gas meters are usually simply adaptations of the gasometer principle modified so as to act continuously. They are of two distinct classes, wet and dry.

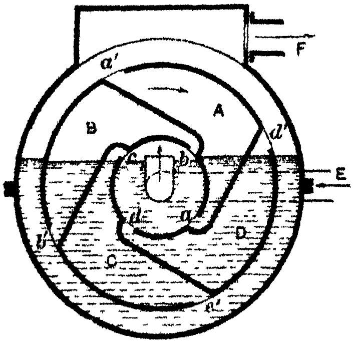

FIG. 324. — SIMPLE FORM OF WET GAS METER.

Wet Meters. — The method of operation of a meter of this type is best explained by the conventional sketch, Fig. 324,* which shows a cross section. The meter consists of a horizontal drum which rotates in a stationary drum and is divided into four compartments, *A, B, C, D.* The lower part of the rotating drum dips into water which opens and closes the slots or ports *abcd,* by which the gas coming through the central pipe *E* enters the respective compartments, and also controls the outlet ports *a'b'c'd'* through which the gas leaves the compartments and reaches the discharge side of the meter at *F*. With the rotation of the drum as shown by the arrow, it will be seen that *B* is filling with gas, while the filling of *C* just starts. *A* is discharging, while the discharge of *D* is just about completed. No gas can go through the meter without registration, because in no compartment will the in- and outlet ports be open at the same time. Rotation is produced by the fact that on account of the slightly greater pressure at the inlet, the water level will be slightly higher on the right side of the meter. Hence this half is continuously heavier than the left side and descends.

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* See Gramberg, *Technische Messungen,* p 121

Such a meter is not used much in practice, because it is not easy to make the ports at the center and in the side wall of the drum of sufficient size. In many constructions, therefore, the ports are placed in the end walls, the inlet ports at one end, the outlet ports at the other. The compartment walls are spirally twisted, because it is necessary to offset the ports about the same as in Fig. 324. The gas then passes nearly axially instead of radially through the meter. A study of Fig. 325, in which the cylindrical side wall of the drum has been removed, may make the construction clear.

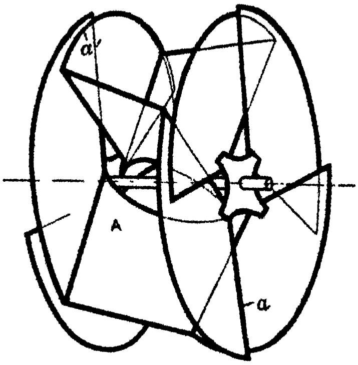

FIG 325.—ORDINARY CONSTRUCTION OF WET GAS METER DRUM.

The accurate indication of wet meters depends upon the height of the water level in the chambers, since the capacity of the latter varies with a drop or rise in the water level. It is therefore essential that this be checked from time to time, because the gas passing will tend to saturate itself and lower the level. Meters constructed for experimental purposes are usually furnished with a fill- and an overflow opening. Through the former water is fed into the housing until it shows at the latter opening. The meter is then allowed to drain out until the flow of water through the overflow ceases, and both openings are then closed. Large meters, such as station meters, are often furnished with continuous water circulation to maintain the proper level. Any wet meter to be used for accurate work should be supplied with spirit level, or levels, so that it may be set true every time it is used.

As in the case of gasometers, the observations required, besides the movement of the drum, which is usually mechanically recorded on dials suitably graduated, are pressure and temperature. The pressure determination should be made in the filling chamber near the end of the filling operation, while the temperature should be that of the gas near the outlet. It is desirable to have gas and water temperature as close together as possible, and to have the water saturated with the gas.

Dry Gas Meters. —The dry gas meter possesses the advantages of not being affected by frost nor of increasing the amount of moisture in the gas. It is made in various forms, and generally consists of two chambers separated from each other by partitions. Each chamber is divided into two parts by a flexible partition which moves backwards and forwards, and actuates the recording mechanism as the gas flows in or out. This motion is regulated by slide valves somewhat similar to those of a steam engine. A type example of such a meter is illustrated in Figs. 326 to 328.* The meter casing is divided into two exterior measuring chambers *PP* by a vertical partition. On each side of this partition there are located two interior measuring chambers *QQ*, which consist of two movable disks *DD* and flexible shells. In Fig. 328, the left-hand side shows the interior measuring chamber exhausted, while at the right the chamber *Q* is filled. Gas is admitted from the inlet pipe *A* to the measuring chambers on the same side of the central partition, that is, to *P* and *Q* alternately, by means of simple slide valves *V*. As *Q* fills, a corresponding quantity of gas is displaced out of *P*, flows through the center port of valve *V* and finds its way into passages *E* and into the discharge pipe *B*. Guides *HH* are provided to preserve parallel motion of the disks *D*. The motion of the latter is transmitted through *G* and rods *R* to the crank mechanism *KKN*, which turns a short vertical spindle. The length of the crank, in order to regulate the disk travel, may be changed by adjusting the position of the pin *N* in the slot shown. The motion of the spindle is transmitted to the

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* Webber, Town Gas and its Uses.

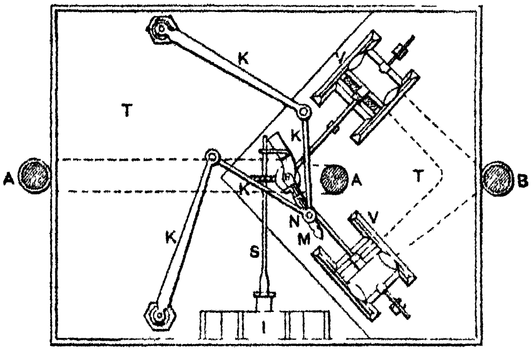
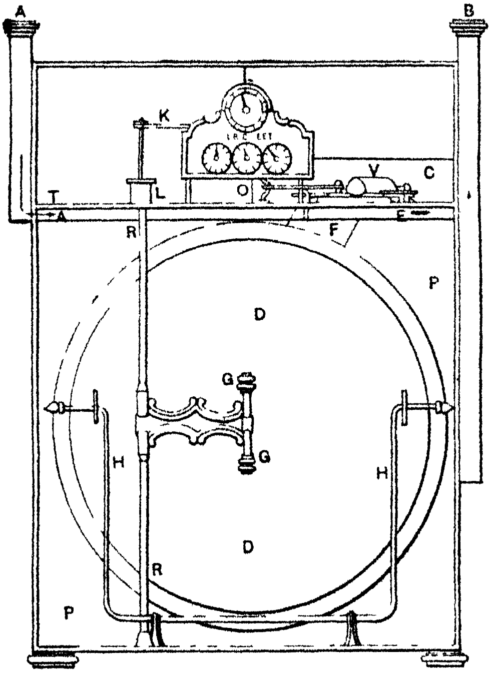


FIG. 326.



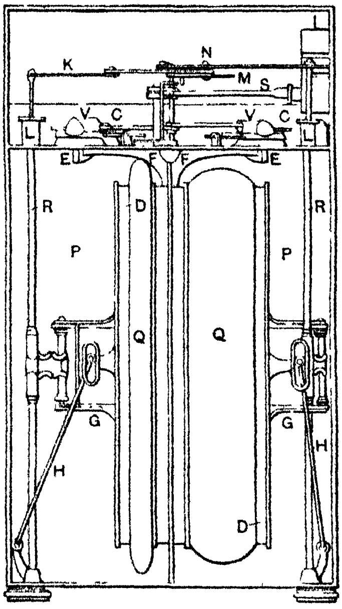


FIG. 327. FIG. 328.

FIGS 326–328.—TYPE OF DRY GAS METER.

index by wheel and shaft *S*. The crank *O*, see Fig. 327, operates the slide valves.

227. Dry Meters of Large Capacity. — The ordinary gas meter, whether wet or dry, soon reaches great proportions when it is attempted to handle large quantities of gas. To obtain large capacity with moderate size, several modifications of the dry meter have been brought out. Fig. 329 shows a rotary or turbine meter, the operation of which is plain from the cut. This type of meter has a minimum limit, like the water meter of similar construction, in that it requires a certain velocity of flow to overcome the frictional resistance of the mechanism. Such a meter gives nearly unrestricted passage and is consequently of large capacity.

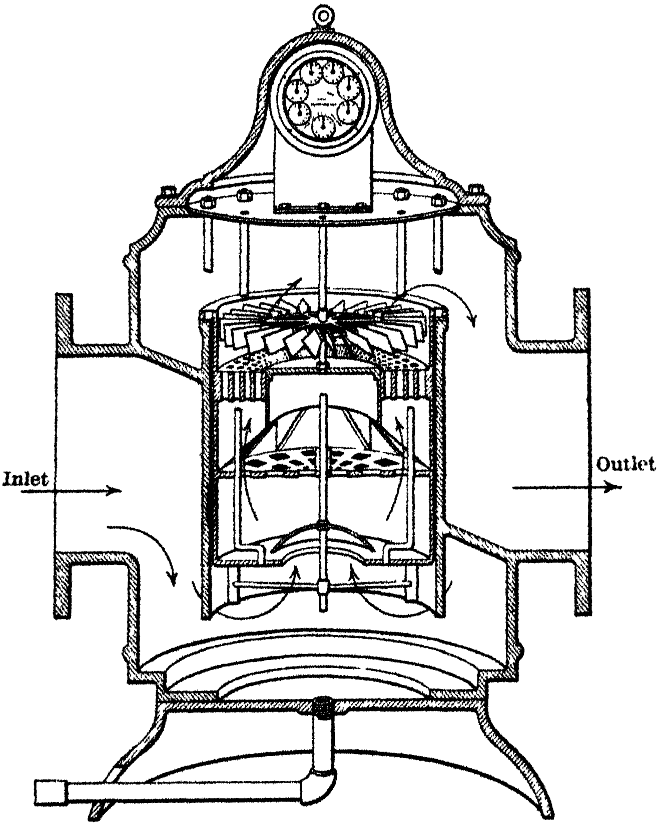


Fig. 329.—Rotary or Turbine Gas Meter.

Another method of reaching the same end is to use the shunt principle, that is, to actually meter only a definite proportional part of the total quantity flowing. Such a construction really consists of two distinct parts; a meter, either wet or dry, but usually the latter, is placed in the smaller shunt pipe, while the main pipe contains a pressure regulating mechanism which is designed to maintain a definite proportion between the volumes passing the two pipes for widely varying inlet pressures. The accuracy of the meter depends directly upon how well the regulating mechanism performs its functions of maintaining proportionality.

228. Calibration of Meters and the Relative Accuracy of Wet and Dry Meters. — It should be made a general rule that no meter be used for experimental work unless it be calibrated. This holds especially for dry meters, as this class is usually inferior to wet meters as far as accuracy is concerned. The only way to calibrate is to send or draw through the meter a definite volume of gas for which the pressure and temperature are known. In every case a calibration curve covering a range of rates of flow should be determined, and to prevent unnecessary computations it is well to construct this curve for standard conditions, which for gas work are fast becoming fixed at 60° F. and 14.7 pounds pressure.

For small meters, such as the Junker gas calorimeter meter, a bottle holding several liters may be conveniently used for calibration, the air or gas being displaced and sent through the meter at different rates by water displacement. For larger meters the meter prover will have to be used.

229. Gas-meter Capacities. — Gas meters are often rated according to the number of gas burners they will supply, a burner being rated at somewhat over 5 cubic feet of gas per hour. Thus, a 10-light meter is good for a normal capacity of about 50 cubic feet per hour, a 200-light meter will have a capacity of about 1000 cubic feet per hour. A dry meter of the latter capacity would call for about a 2-inch pipe connection.

Meters of the rotary type are made in capacities from about 1500 to 100,000 cubic feet per hour. For the latter meter, the pipe connection would be about 30 inches, for the former about 3 inches. Proportional meters can be had with capacities up to 200,000 cubic feet per hour.

230. Reservoir Method of Measuring Air or Other Gases. — This method is somewhat similar to the gasometer method except that the volume instead of the pressure is kept constant. It is sometimes used in measuring the discharge of air compressors. For this purpose the compressor is made to deliver against any desired pressure, which is kept constant by a regulating valve, into a tank or receiver until the pressure in the tank approaches the desired delivery pressure. The readings necessary are the following: p_1 and t_1 in the tank at the beginning of a trial, p_2 and t_2 at the end of the trial and time required to pump from the pressure p_1 to the pressure p_2. A relief valve is provided between tank and regulating valve which serves to again reduce the pressure in the tank and through which the compressor may also discharge between trials. Concerning the pressure determination, it is well to state that if a gauge is used p_1 should be taken at some pressure above the atmosphere, as the gauge indications are generally very uncertain near the zero of the scale. The bad feature of the receiver method of measuring air consists in the determination of the average air temperature. The temperature is not likely to remain constant during a given trial, but changes, due to the different temperature of the incoming air, to the heat developed by compression in the receiver itself, and to the loss by radiation. If possible, the higher pressure should be maintained in the tank until the final temperature is determined, which is not particularly easy where the volume is large. If the latter method is adopted, a tight tank is an absolute requirement.

As far as the computations are concerned, the result will be expressed in weight of air, since the volume V of the tank in cubic feet is a constant. For the condition at the beginning, the tank contains a weight of air equal to

$$W_1 = \frac{VP_1}{53.3\,T_1} \text{ pounds,}$$

53.3 being the value of the constant R for air and P the absolute pressure in pounds per square foot.

For the end condition we have similarly

$$W_2 = \frac{VP_2}{53.3\,T_2} \text{ pounds.}$$

The weight added in the given time is therefore

$$W = W_2 - W_1 = \frac{V}{53.3}\left(\frac{P_2}{T_2} - \frac{P_1}{T_1}\right) \text{ pounds.} \qquad (43)$$

This weight of air can then easily be converted to cubic feet of compressor discharge when the pressure and temperature conditions are given.

In case the gas consumption of any given machine or apparatus, as for instance a gas engine, is desired, the above described receiver method may be inverted; that is, the tank is pumped up to a high pressure and the gas is then throttled to the desired pressure. The measurements and computations are the same as above. The method is likely to give better results because the temperature determination can be made in the outlet pipe with more certainty of obtaining an average.

IV. MEASUREMENT OF GAS BY DETERMINATION OF AVERAGE VELOCITY.

The principal instruments used for this purpose are anemometers, Pitot tubes, and Venturi meters.

231. **Anemometers,** as far as the principle of their action is concerned, are very similar to current meters, see Art. 210, p. 402, and what is stated there concerning accuracy applies equally well to these instruments. Anemometer wheels are of course built as light as possible consistent with stiffness, mounted in bearings that can be nicely adjusted, and are very carefully balanced.

A form shown in Fig. 330 with flat vanes, and with the dial arranged in the center as shown, or on top of the case in various positions, is much used as a portable instrument.

The dial mechanism of the anemometer can be started or stopped by a trip arranged convenient to the operator; in some instances the dial mechanism is operated by an electric current similar to that described in connection with the current meter. They can also be made self-recording by

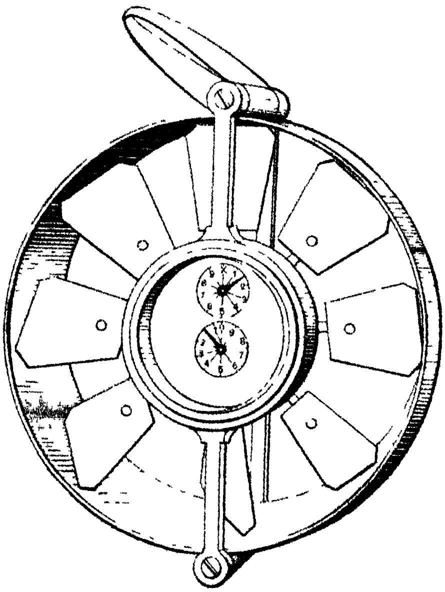


FIG. 330. — ANEMOMETER.

attaching clockwork carrying an endless paper strip which is moved under a pencil operated by the anemometer mechanism.

Robinson's anemometer, which consists of hemispherical cups revolving around a vertical axis, is much used for meteorological observations. This type of instrument can be used for higher velocities than the other; it is good for about 200 ft. per sec., while the other often fails at about 100 ft. per sec.

Calibration of Anemometers. — Anemometers are calibrated by moving them at a constant velocity through still air and noting the readings on the dials for various rates of movement. This is usually done by mounting the anemometer rigidly on a long horizontal arm which can be rotated about a vertical axis at a constant speed. The distance moved by the anemometer in a given time is computed from the known distance to the axis and the number of revolutions per minute; from these data the velocity is computed.

In performing this experiment care must be taken that the axis of the anemometer is at right angles to the rotating arm. Readings should be taken at various speeds, since the correction is seldom either a constant quantity or one directly dependent on the velocity.

Anemometers can also be calibrated by measuring the quantity of air flowing through a conduit with some other apparatus and comparing the anemometer reading in the same conduit with the velocity thus determined. The test should be made with different quantities so as to calibrate the anemometer over its entire range. Sometimes such a test is made by placing the anemometer in a pipe which tightly fits the casing, thus causing all of the air measured to pass through it, computing the actual velocity from volume and cross section. It will in general be found, on comparing the result with that obtained for the same *actual* velocity when swinging the anemometer through still air, that the results do not check, but that the velocity shown in the latter case, where the air may pass around the outside of the anemometer case, is less than that given by the former method. The difference may amount to 20 per cent, and it becomes a question to decide which one of the calibration results is the one to use. Evidently the conditions of calibration are not the same. That being the case, it is next evident that the conditions under which the anemometer is used should determine the method of calibration. If, for instance, the velocity of the wind is to be found, the anemometer should be calibrated by swinging it. On the other hand, if the cross section of the pipe or duct in which gas velocity is measured is not very great as compared with the size of the anemometer, evidently the second method of calibration is to be preferred. But between these extremes there are a variety of ntermediate cases, and this fact is what makes the anemometer in some instances an unreliable instrument.

232. Precautions Necessary in the Use of Anemometers. — These nstruments possess upper and lower velocity limits beyond which they should not be used. When the velocity is very low the friction of the instrument is relatively so great as to make it unreliable or otally inoperative, as is the case with current meters. When the velocity is very high an anemometer built for ordinary service is apt o be bent out of shape, due to its light construction, and the friction oss is often considerably increased.

Due to the delicate construction of most of these instruments they require careful handling, and they should be frequently calibrated o guard against changes of the calibration constants due to bending of the vanes, wear of parts, etc.

233. Measurement of Gas by Means of Pitot Tubes. — The device lescribed in Arts. 213 to 215 for use with water may also be used for neasuring the velocity of gases, and the statements there made concerning accuracy, etc., apply in general also to this case. The velocity may be computed from the formula

$$V = C\sqrt{2gH} \qquad (44)$$

n which C is an experimentally determined constant and H is the velocity head. H depends upon the gas used, and care must be aken to compute it properly from the velocity head as measured. If water is used as the manometer liquid, let h = the velocity head n water inches. Then $H = hr$, where r is the number of feet of gas which will give the same pressure as 1 inch of water. In general, or any manometer liquid $H = kh'r$, where r is as above, h' = velocity head as measured in inches, and k is the density of the nanometer liquid as compared with water. C should be close to 1.0, if all conditions are properly adjusted.

For a temperature of 60° F. and a barometric pressure of 29.92

inches Hg, 1 inch of water column will balance 68 feet of dry air, or $r = 68$.

It should be noted that for ordinary gas velocities the actual manometer reading may be very small, so that it is very difficult to guard against appreciable errors in the calculated velocities. The best that can be done is to use some form of differential manometer so as to make the manometer reading as reliable as possible; see Art. 91, p. 174.

The density of gas increases directly with the absolute pressure and inversely as the absolute temperature; it varies also with moisture, so that corrections are required for pressure, temperature, and the amount of moisture.

When the precautions and directions laid down for the use of the Pitot tube with water are observed, the instrument will give satisfactory results also for gases. A thorough discussion of the use of the Pitot tube for the measurement of gases flowing in pipes is given by R. Threlfall in a paper before the British Institution of Mechanical Engineers in 1904.

234. Measurement of Gas by Venturi Meter. — Recent experiments show the Venturi meter to be a satisfactory instrument for the measurement of gases.* For the theory for this instrument see Art. 195, p. 369.

The formula for velocity in the throat of such a meter is

$$V_2 = C\left(\frac{P_1}{\delta_1}\right)^{\frac{1}{2}} \left(\frac{2\,g\gamma}{\gamma - 1}\right)^{\frac{1}{2}} \left(\frac{1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}}}{1 - \left(\frac{F_2}{F_1}\right)^2 \left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}}}\right)^{\frac{1}{2}} \qquad (45)$$

in which

Subscripts 1 refer to the upstream end of meter,

Subscripts 2 refer to the throat of meter,

P is the pressure in pounds per square foot,

δ is the density of the gas being metered, pounds per cubic foot,

F is the area in square feet,

---

* See paper "The Flow of Fluids in a Venturi Tube," by F. P. Coleman, Trans. A. S. M. E., Vol. 28, p. 483.

γ is the ratio of specific heats, $\frac{C_p}{C_v}$, and

C is an experimentally determined constant.

The value of C is not as yet well determined for widely differing conditions, but it seems to be practically constant over a wide range of throat velocities and to be near unity. In any case, a meter once calibrated over a sufficient range can be used without sensible error over the same range with little danger of the constant changing, unless the gas carries foreign matter such as dust or tar in suspension. In such cases periodic cleaning must be resorted to.

V. MEASUREMENT OF GAS BY CALORIMETRY.

235. **Methods Used.** — This method offers a very convenient and comparatively simple means of measuring gas. It is applicable to the measurement of both large and small quantities, but is apt to give erratic results when the quantities become so small that radiation may cause an appreciable error. For large quantities it is one of the best.

In principle it is simple: a measured quantity of heat is added to the gas flowing through a conduit and the rise of temperature is measured; then

$$H - r = WC_p (T' - T), \tag{46}$$

in which

H = heat supplied, in a given time,
r = radiation loss in same time,
W = weight of gas passed in same time,
C_p = specific heat of gas at constant pressure,
T' = final temperature of gas,
T = initial temperature of gas.

From this

$$W = \frac{H - r}{C_p (T' - T)}$$

and if radiation be neglected

$$W = \frac{H}{C_p (T' - T)}.$$

A method of making the measurements above outlined is illustrated in Fig. 331, in which the gas enters the pipe or channel at A

and is discharged at *D*. Means for heating, which may be either a steam or electric radiator, is to be supplied. If a steam radiator is used, the heat discharged is computed from measurements of the weight and temperature of the condensed steam, the heat entering from measurements of pressure, quality, and weight by methods already explained. The heat taken up by the air is the difference of that entering and discharged. If an electric heater is used, the electric energy disappearing is measured and reduced by computation to heat-units. The means for heating should be of such form as to heat the air uniformly.

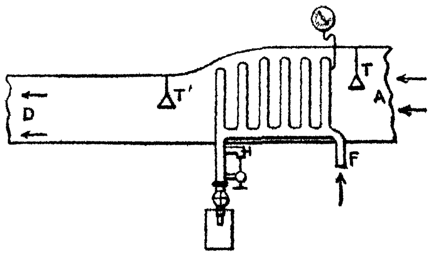


FIG. 331.—CALORIMETRIC METHOD OF MEASURING GASES (Steam Heater).

The temperature of the entering and discharged air should be taken at sufficiently numerous points in the cross section to make the average results accurate, and the thermometers should be protected from radiant heat. The average temperature should also be measured at the section where the velocity is to be computed. It may be desirable, in case extreme accuracy is required, to compute the weight of moisture in the air from observations with the dry- and wet-bulb thermometer.

Very satisfactory results are obtained by the use of resistance thermometers, because the wire forming the resistances can be strung on frames or insulators in such a way as to be well distributed over the entire section of the channel, giving a good average temperature reading. An electric meter constructed on these principles is now in the market.*

As the actual rise of temperature in practical cases is small, radiation may in general be disregarded. In case it is desirable to allow for it an approximate value may be obtained by stopping the flow of gas and measuring the heat input necessary to maintain the heating chambers at a temperature about halfway between the values of entering and leaving temperatures when the gas is flowing.

* See an article on the Thomas electric gas meter in *Machinery* [illegible]

MEASUREMENT OF VAPORS.

236. General Considerations. — In general, vapors may be measured by any of the means used in the measurement of gases, but when dealing with vapors wet, dry and saturated, or only slightly superheated, many of these methods give very questionable results. The laws of vapors slightly superheated, that is relatively near the dry, saturated condition, are not as yet well known. During the past few years much work has been done in this region in the case of water vapor, but even here considerable doubt still exists. The more the vapor is superheated, that is, the further it is removed from the condition of saturation, the more nearly does it obey the laws of perfect gases and therefore the more accurate do the methods of gas measurement become.

For vapors which are near the saturated condition, or are actually saturated, it is best, when possible, to base all measurements upon the weight or volume of the liquid before vaporization or after condensation. This is the most common method of measuring steam and is undoubtedly the best.

The various methods in more or less common use for the measurement of vapors may be grouped as follows:

I. Determination of the volume or weight of liquid before vaporization or after condensation.

II. Measurement by means of orifices.

III. Measurement of the average velocity of flow in a pipe or conduit.

IV. Calorimetric measurement.

I. DETERMINATION OF WEIGHT OF LIQUID.

237. Methods Used. — For this purpose any of the methods given for the measurement of liquid may be used. The most common methods are direct weighing or the use of a calibrated meter. In engineering the vapor to be measured is generally steam, and it is customary to measure either the boiler feed water or the liquid resulting from condensation in some apparatus. It should be observed that in either case there are possibilities of considerable leakage, and unless these are guarded against or the leakage actually measured, errors of unknown magnitude may occur.

II. MEASUREMENT BY MEANS OF ORIFICES.

238. General Case. Theoretical Velocity Attained. — If it is again assumed that the flow is adiabatic, and that there is no friction, a brief consideration will show that the kinetic energy that the vapor has gained in passing the orifice must be exactly equal to the difference in the heat content of the vapor under the initial and final conditions.*

Case of Saturated Vapor.

The equation for velocity can be developed by making the proper substitution for the intrinsic energy in equation (31), p. 420. For a mixture of a liquid and its vapor the intrinsic energy generally is

$$E = \frac{1}{A}(x\rho + q),$$

in which $A = \frac{1}{778}$,

x = quality of the vapor mixture,
ρ = internal latent heat,
q = heat of the liquid.

As in the case of gases, we then obtain

$$\frac{V_2^2}{2g} = \frac{1}{A}(x_1\rho_1 + q_1 - x_2\rho_2 - q_2) + P_1v_1 - P_2v_2. \qquad (47)$$

Now in general,

$$v = xu + (1 - x)\sigma,$$

in which v = specific volume of the mixture of liquid and vapor,
u = increase in volume during vaporization,
σ = specific volume of the liquid.

The factor $(1 - x)\sigma$ is so small that for ordinary engineering work it may be neglected, and substituting for v_1 and v_2 in equation (47), we shall have

$$\frac{AV_2^2}{2g} = x_1\rho_1 - q_1 - x_2\rho_2 - q_2 + AP_1x_1u_1 - AP_2x_2u_2. \qquad (48)$$

To simplify this equation, write $r = \rho + APu$, whence

$$\frac{AV_2^2}{2g} = x_1r_1 + q_1 - x_2r_2 - q_2, \qquad (48a)$$

from which finally

$$V_2 = \sqrt{\frac{2g}{A}(x_1r_1 + q_1 - x_2r_2 - q_2)} \text{ ft. per sec.} \qquad (49)$$

---

* Strictly, this statement is not accurate on account of the fact that the APu values are not the same for the initial and final conditions.

In this formula the terms r_1 and q_1 are known from the steam table for the pressure P_1, and x_1 can easily be found by calorimeter. The terms r_2 and q_2 become known as soon as the final pressure P_2 in the orifice or nozzle (which is not necessarily that of the atmosphere into which the vapor may be discharging) is determined (see Art. 221). x_2, the quality at the pressure P_2, may then be found from the entropy equation

$$\frac{x_1 r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2, \qquad (50)$$

in which the term $\frac{xr}{T}$ represents the entropy of the vapor and θ that of the liquid. P_1, P_2, and x_1 being known, all the terms of this equation except x_2 may be obtained from steam tables (see Appendix).

Case of Vapor Initially Superheated.

(*a*) Vapor wet in final condition. Usual case. The simplest way of taking care of this is to add some term to equation (48a) to allow for the heat represented by the degree of superheat. Thus equation (48a) may be modified to read

$$\frac{AV_2^2}{2g} = \int_{t_1}^{t_3} c\,dt + r_1 + q_1 - x_2 r_2 - q_2. \qquad (51)$$

Here, as in the case of saturated steam, the value of x_2 must be found, which can be done from

$$C_D \log_e \frac{T_3}{T_1} + \frac{r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2, \qquad (52)$$

where T_3 = absolute temperature of the superheated vapor,

T_1 and T_2 are the saturation temperatures of the vapor at pressures P_1 and P_2 respectively,

and C_D is the mean specific heat at constant pressure of the superheated vapor in the temperature interval $T_3 - T_1$.

(*b*) Vapor remaining superheated throughout. This case is not of much practical importance. The equations applying to it may be easily developed in case of necessity from equations (50) and (51) by certain modifications.

239. Method of Determining Theoretical Velocity for Steam from the Heat Chart, Fig. 245.—A quick method of determining V_2 is available by the use of the diagram Fig. 245. From the point on the diagram corresponding to the initial condition of the steam move down a vertical (isentropic) line to a point of intersection with the pressure P_2, existing in the smallest section F in the nozzle or other orifice. This corresponds to the adiabatic change. The first point on the diagram gives the heat content of the steam in the beginning, the last the heat content after expansion in the nozzle. Calling the difference B, we may compute the velocity from

$$V_2 = \sqrt{\frac{2g}{A}B} \text{ ft. per sec.} \qquad (53)$$

240. Value of the Pressure P_2. — As in the case of the gases the value of the pressure P_2 is not necessarily that of the pressure in the discharge space, there being critical pressures and critical ratios which appear to change slightly with the shape of the orifice and with the condition of the vapor as to quality.

None of the equations so far given for steam will serve to bring out this critical data, as none of them involve the pressure ratios. Equations exactly similar to those for gases have, however, been developed by Zeuner and others. Zeuner gives the following expression for the theoretical velocity:

$$V_2 = \sqrt{2gP_1v_1\frac{\gamma}{\gamma-1}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right]} \text{ ft. per sec.} \qquad (54)$$

This value for V_2 becomes a maximum when the pressure in the nozzle is

$$P_2 = P_c = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} P_1. \qquad (55)$$

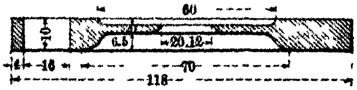
For saturated steam, with x not less than .7, we may write

$\gamma = 1.035 + .1\,x,$
so that for $x = 1.0, \quad \gamma = 1.135$
and for $x = .7, \quad \gamma = 1.105.$

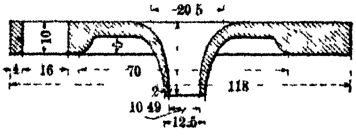
This about covers the field of saturated steam as commercially used, and for these values of γ, $P_c = .578\,P_1$ in the case of the higher quality, and $= .583\,P_1$ in the case of the lower. Evidently the critical value $P_c = P_2 = .58\,P_1$ fits every ordinary case of saturated steam.

For superheated steam γ = approximately 1.29, for which P_c would be = .55 P_1. But in the ordinary case the superheated steam expanding becomes wet and changes its value of γ to near those first quoted. The case is complicated, the value of P_c then being somewhere between .55 and .58 P_1.

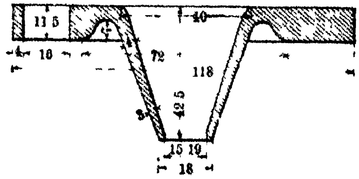
Orifice in Thin Plate



A Nozzle 10.49 Mm. Diam at 15°C.



B Nozzle 15.19 Mm. Diam at 15°C.



C Nozzle 24.20 Mm Diam at 15°C

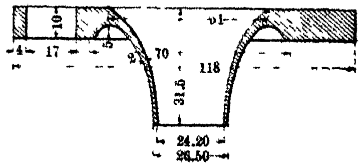


FIG 332.

To apply equations (49) and (51) in any practical case, it therefore again becomes necessary, as for a gas, to check the pressure ratio $\frac{\text{Pressure in Discharge Space}}{\text{Pressure in Reservoir}}$. If this is greater than .58 (or .55 for superheated steam), r_2 and q_2 apply to the pressure in the discharge space, and x_2 may then be found from equations (50) or (52). If, on the other hand, the ratio is less than .58 (or .55), the pressure for which r_2 and q_2 (and also x_2) are found will be for the critical pressure = .58 (or .55) P_1.

241. Weight of Vapor Discharged. — If F is the area in sq. ft. of the section of the nozzle at which V_2 and P_2 are established (in convergent nozzles and bell-mouth orifices the smallest section), and δ is the density (pounds per cubic foot of the mixture of liquid and vapor, or vapor alone if superheated), the theoretical discharge weight will be

$$W = F\delta V_2 \text{ lbs. per sec.} \tag{56}$$

In any actual case this is modified by a coefficient of discharge C, for which values are given below, so that

$$W = CF\delta V_2 \text{ lbs. per sec.} \tag{57}$$

242. Value of the Coefficient of Discharge C. — Comparatively few experiments have been made on orifices or nozzles properly shaped to give the maximum discharge. Perhaps the most extensive ones made on steam are those of Rateau* and Gutermuth.†

The dimensions of the orifices and nozzles used by Rateau are given in Fig. 332. These have been left in millimeters, from which they may be easily reduced to inches by dividing by 25.4. The results obtained plotted as the ratio of the discharge W actually obtained to the theoretical maximum discharge W_m, are given in the curves, Fig. 333. The initial pressures used went up to about 145 pounds per square inch for nozzles A and B, to about 85 pounds for nozzle C, and to 75 pounds for the thin-plate orifice. The curves show that for a properly designed mouth-piece or convergent nozzle, the constant C is again slightly below 1.0, as for air. The curve A–B is the curve of theoretical discharge, while C–D is the average of the figures

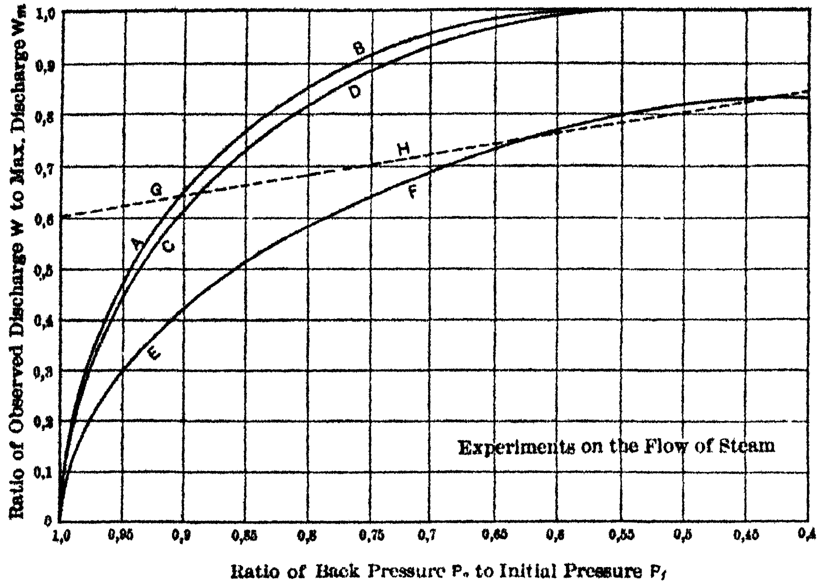


FIG. 333.—VALUES OF DISCHARGE COEFFICIENT C FOR FLOW OF STEAM THROUGH NOZZLES.

Curve A–B is the *theoretical* discharge, curve C–D the average *actual* discharge for the three nozzles A, B, and C, Fig. 332. The discharge coefficient C is then the ratio of $\frac{\text{ordinate of } C\text{–}D}{\text{ordinate of } A\text{–}B}$.

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* Experimental Researches on the Flow of Steam through Nozzles and Orifices, A. Rateau, translated from the French by H. B. Brydon.

† Gutermuth, Experiments on the Flow of Steam through Orifices, *Zeitschrift d. V. d. I.*, Jan. 16, 1904.

obtained for all three nozzles. Taking the ratio of the ordinates of these curves gives the coefficient C. This increases from .94 at $\frac{P_2}{P_1}$ = about .96 to .967 at $\frac{P_2}{P_1} = .8$, to .974 at $\frac{P_2}{P_1} = .7$, and has practically the value 1.0 when the critical ratio $\frac{P_2}{P_1} = .57$ is reached.

The results for the thin-plate orifice are shown in curve E–F. In this case, again taking the constant equal to the ratio of ordinates of E–F to ordinates of A–B, the variation is shown by the practically straight line G–H. The coefficient starts with a value of .622 at $\frac{P_2}{P_1} = .9$ and increases to .770 at $\frac{P_2}{P_1} = .6$. It still seems to increase after the critical ratio $\frac{P_2}{P_1} = .58$ is passed.

A close analysis of Rateau's work will show that it is subject to several cumulative errors, especially for pressure ratios around the critical, so that the values of C may be as much as 2 per cent too high. The inaccuracies are partly due to new steam-table data which was not available at the time the work was done.

Gutermuth's work was done with a view to determining the conditions controlling the flow of steam through valve ports of steam engines, and to that end, besides investigating circular, sharp-edged, and bell-mouth orifices, he also tested rectangular orifices with curved extensions approximating the shape of valve ports. Only the results on the circular orifices are of interest here.

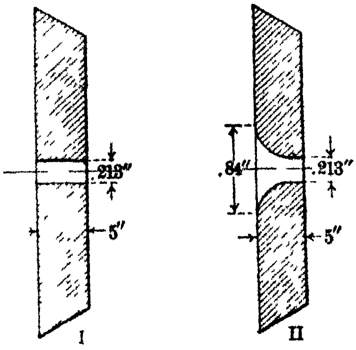

FIG. 334.

Fig. 334 shows the dimensions, estimated from the original cuts, of the circular orifices used.

Form II gave a constant C very close to 1.0, as might be expected. This orifice practically gave the theoretical discharge as computed by Zenner's equation and equation (57), taking $\gamma = 1.135$. Form I, not a true thin-plate orifice, gave the following results, the initial pressure being about 130 pounds per square inch absolute.

For $\frac{P_2}{P_1}$ =	.977	.944	.888	.777	.666	.611	.577
C =	.705	.78	.805	.846	.855	.868	.88

These values are throughout greater than those found by Rateau for a thin-plate orifice, as might be expected.

In applying the orifice method of measuring steam it therefore seems better to use a well-rounded orifice than an orifice in a thin plate, on account of the greater certainty in the coefficient and its smaller variation.

243. Approximate Formulæ for the Flow of Steam through Orifices. —If in equation (54) we substitute $g = 32.2$, $\gamma = 1.135$, and $\frac{P_2}{P_1} = .58$, and $P_1 = 144\,p_1$, we obtain

$$V_2 = 70\sqrt{p_1 v_1} \text{ ft. per sec.,} \qquad (58)$$

and in theory this velocity will be constant as long as the pressure in the discharge space is less than .58 P_1. Now for saturated steam

$$p_1 v_1^{1.065} = \text{constant} = \text{approx. } 485, \qquad (59)$$

where p_1 is pounds per square inch, and v_1 = specific volume in cubic feet.

From this
$$v_1 = \frac{332.6}{p_1^{.939}},$$

and substituting this value in equation (58), we have

$$V_2 = 1274\sqrt{p_1^{.061}} \text{ ft. per. sec.} \qquad (60)$$

The weight discharged will then be

$$W = CF\,\delta\,V_2 = 1274\,CF\,\delta\sqrt{p_1^{.061}} \text{ lbs. per sec.} \qquad (61)$$

δ is the density of the vapor at the lower pressure P_2, and assuming that the vapor is still saturated at that pressure, we may write

$$\delta = \frac{1}{v_2} = \frac{p_2^{.939}}{332.6} = \frac{(.58\,p_1)^{.939}}{332.6}.$$

Substitute this value of δ in equation (61) and we obtain

$$W = 2.3\,CFp_1^{.97} \text{ lbs. per sec.} \qquad (62)$$

For an *orifice with well-rounded entrance,* put $C = 1.0$, and if F is expressed in square inches instead of square feet, we will finally have for such an orifice

$$W = .016\,Fp_1^{.97} \text{ lbs. per sec.} \qquad (63)$$

This is Grashof's formula, and the dimensions of the equation are: W, in pounds per second; F, in square inches of smallest area; and p_1, the initial pressure in pounds per square inch.

Grashof's equation may be written

$$W = \frac{F p_1^{.97}}{62.5} \text{ lbs. per sec.} \qquad (64)$$

An equation of very similar form was developed by Rankine upon the basis of work done by R. D. Napier. He gives

$$W = \frac{F p_1}{70}, \qquad (65)$$

and this equation is known as *Napier's formula.*

When the pressure P_2 is greater than .58 P_1, the flow, according to Rankine, may be approximately expressed by

$$W = F \frac{p_2}{42} \left\{ \frac{3 (p_1 - p_2)}{2 p_2} \right\}^{\frac{1}{2}} \text{ lbs. per sec.} \qquad (66)$$

Note that the pressures in equation (66) are in pounds per square inch.

The error of either one of the approximate equations is likely to be ± 2 per cent.

III MEASUREMENT OF VAPOR BY DETERMINING AVERAGE VELOCITY. STEAM METERS

244. The instruments which may be used for this purpose are mainly Pitot and Venturi tubes. The method of applying these to gases has been outlined in Arts. 233 and 234. The laws are practically the same for vapors.

245. Steam Meters. — When the readings of either one of the above instruments are made continuous in some manner, we have what is known as a steam meter. There are, however, several other fundamental types of steam meters,* and for the purpose of explaining their principle they may be generally classified as

(*a*) Meters employing the Pitot tube principle.

(*b*) Meters employing the Venturi tube principle.

(*c*) Orifice meters.

(*d*) Float meters.

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* Probably the most extensive treatise on steam meters is that given by F. Bendemann in the *Zeitschrift des Vereins deutscher Ingenieure*, Jan. 2 and 23, 1909. To this treatise several of the following cuts are due.

Some of the oldest steam meters employed a wheel and were constructed much like the present-day turbine water meter, but that design has apparently been given up. The number of steam meters on the market to-day is considerable. Their accuracy depends upon how well the various factors entering the problem are taken care of. The makers of the best of them will guarantee an accuracy of ± 3–5 per cent on steady flow. None of these are apparently quite satisfactory on intermittent or pulsating flow. In any case where they must be used with such flow, a special calibration after installation is necessary.

What may be called the "operating" part of most of the steam meters is very simple; the "recording" apparatus is, on the other hand, in many cases quite complex, especially if compensations for pressure and temperature variations are made.

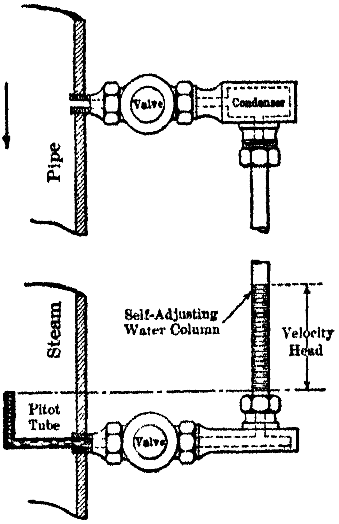

FIG. 335.—BURNHAM STEAM METER.

The quantity of steam flowing through any given main depends upon the function $\sqrt{p_1 - p_2}$ and upon the density δ, which for practical purposes may be expressed by $\sqrt{p_1}$. A meter to be accurate must therefore account for both of these factors, and if the steam should at any time be superheated, a temperature correction also enters. Apparently only one of the commercial meters corrects for temperature, and even in that case the adjustment is made by hand and not automatically.

(*a*) *Pitot Tube Meters.* The simplest form of this meter is apparently that designed by Burnham, Fig. 335. The height of the water column in the glass is a function of the velocity. The position of the impact opening with reference to the cross section of the pipe is evidently of importance. It should in theory be placed at the point of *average* velocity, but the determination of this position in practice is not so easy and would in any case require traversing with various velocities of current. The density factor is apparently not taken care of in the instrument and will have to enter into the computation for weight.

A much more elaborate meter based on this principle is that made by the General Electric Company. Here the static and impact tubes, combined in what is called the nozzle plug, Fig. 336, extend nearly across the diameter of the pipe in which the flow is to be measured. The two tubes leading from this plug are connected to the two unions shown near the top of Fig. 337, which illustrates the recorder used. This consists essentially of two vertical tubes partly filled with mercury, connected across near the bottom by another tube, so that the whole practically forms a U-tube manometer. The entire arrangement is balanced on knife-edges as shown. The nozzle plug tubes are connected to the manometer by flexible tubing, one to each branch of the tube. The excess of pressure on one side causes a part of the mercury to flow over to the side of least pressure, causing the beam to move about the knife-edge until balance is again established. The movement of the beam, which is multiplied by levers, is thus a direct function of the difference in pressure, i.e. of the velocity of the current of steam through the pipe. The rest of the instrument is a clock arrangement for recording the movement of the beam on paper, which is graduated so that the flow can be read directly in pounds per hour. Compensating devices for pressure and temperature are fur-

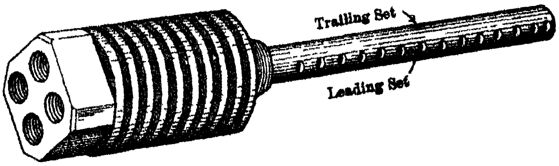

Fig. 336.—Nozzle Plug, G. E. Co's Steam Meter.

nished. Pressure variation is corrected for automatically. A tube of the Bourdon type, influenced by the static pressure at the point where the velocity is being measured, causes a small "correction" weight to shift so as to correct the pencil travel. Temperature is compensated for by hand adjustment of the same weight, according to a special calibration curve sent out for each meter. This meter is also made in two other forms, as a recording meter without the compensating device, for installations where neither pressure nor temperature vary to any great extent, or as a simple indicating meter where no continuous record of flow is necessary.

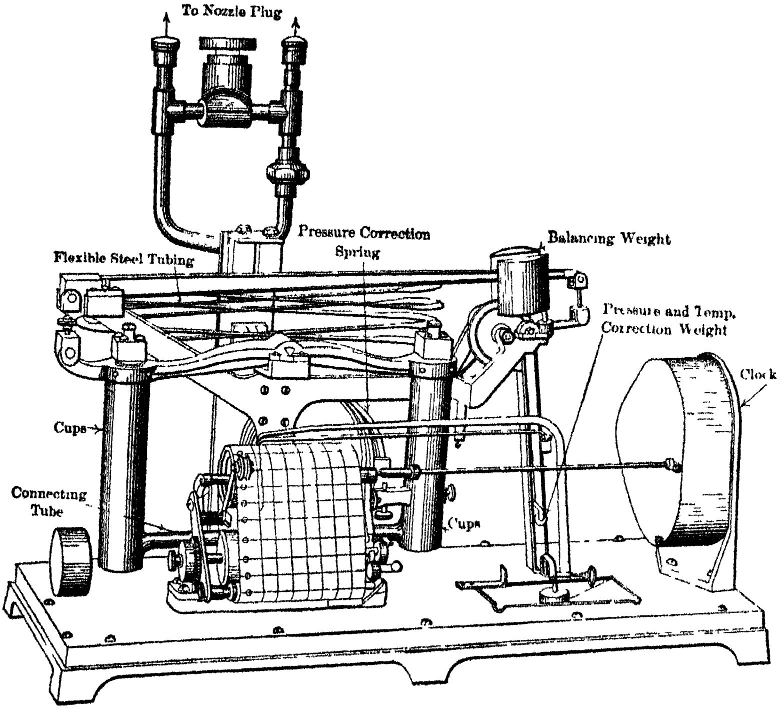

FIG. 337.—RECORDING APPARATUS, G. E. CO'S STEAM METER.

(*b*) *Venturi Steam Meter.* The only meter of this type so far brought out is that made by J. C. Eckardt, Stuttgart-Cannstatt, and is shown in Fig. 338. The Venturi tube is combined with a steam separator, which, by the way, is always good practice in connection with any steam meter. The tube R_1 leads to the high-pressure space, while the pressure existing at the throat is measured by means of the small tube *l* and the larger tube *R*. Both pressures are autographically recorded, the strip of paper showing the pressure difference continuously. The average pressure difference may from this be found by the planimeter. The steam flow is then read off directly from tables furnished with the instrument. The weak point of the meter seems to be the recording apparatus. In the first place the two recording manometers must maintain a fixed adjustment with relation to each other. In the second place, while the distance between the two pressure curves may be 1.5 or 2 inches for maximum flow, as it is in the instrument illustrated, this distance rapidly de-

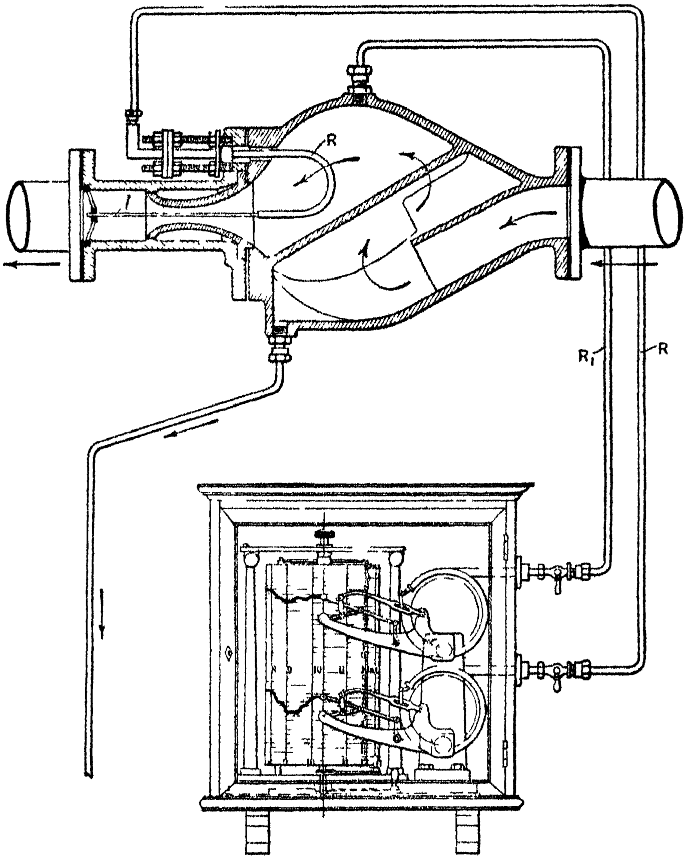

FIG 338 —ECKARDT STEAM METER.

creases, so that it is only a few hundredths when the flow is 10 per cent of the normal. Under these conditions the error of planimetration may be quite serious.

(c) *Orifice Meters.* The principle upon which these meters operate is well shown by Figs. 339 and 340, illustrating the meter made by Hallwachs & Co.* The method of measuring the pressures existing ahead of and behind the orifice is shown in Fig. 340. The indicating apparatus, Fig. 339, consists of a simple form of differential multiplying mercury manometer. The spiral coils in each pressure pipe are put in for the purpose of maintaining the same height of water column on the mercury on each side irrespective of pressure variation. For a constant steam pressure, that is for constant δ, the steam flow-

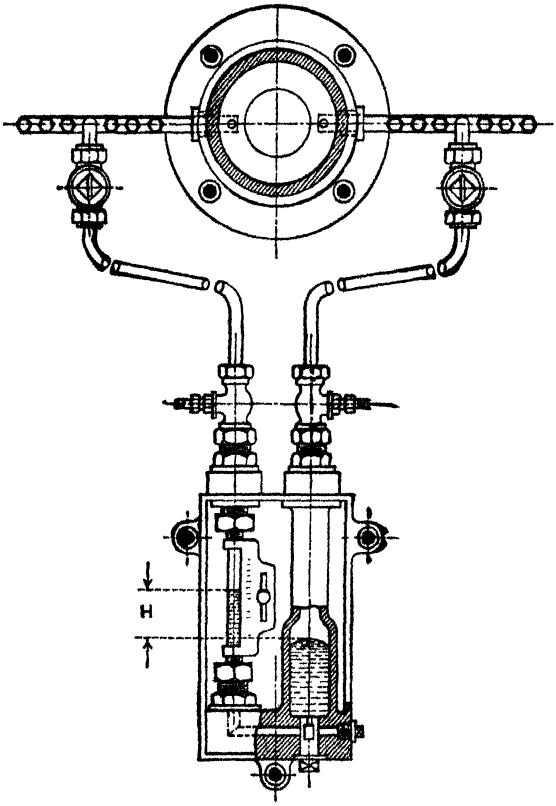

FIG. 339.—HALLWACHS' STEAM METER.

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* Malstatt St.-Johann an der Saar.

ing may be computed from the head H. To obtain the results directly for different steam pressures, various flow scales are sometimes placed on a cylinder which is pivoted next to the manometer, and the flow may be read off directly by bringing the proper scale into position. This instrument has been made recording by electrical means by fusing into the manometer tube metallic contacts at proper intervals. As the mercury reaches each contact in turn, an electric circuit is closed and a line proportional in length to the square root of H is recorded. From this diagram the average $\sqrt{H}$ may then be directly found. The density factor $\sqrt{p_1}$ is apparently not accounted for. This type of meter is otherwise quite reliable and many of them are in use on the continent.

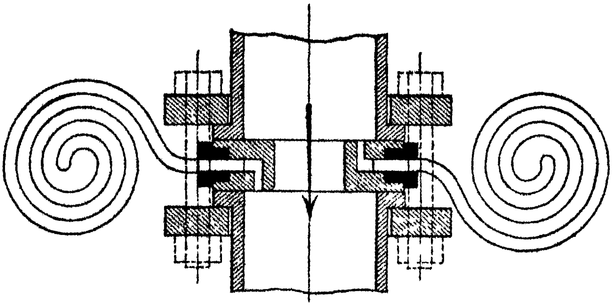

FIG. 340.—ORIFICE, HALLWACHS' STEAM METER.

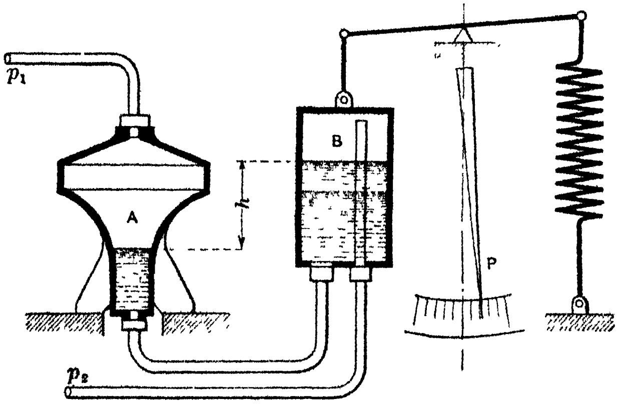

FIG. 341.—CONVENTIONAL SKETCH SHOWING PRINCIPLE OF GEHRE RECORDING APPARATUS FOR STEAM METER.

Several other designs of orifice meters, differing in their recording apparatus, are on the market. One of the most recent and one of the most accurate is that brought out by M. Gehre,* who has turned out several designs within the past few years. The recording apparatus of the last Gehre meter is quite complicated, but the principle underlying it may be explained from Fig. 341. The higher pressure p_1 acts on the surface of the mercury in the stationary vessel *A*. This vessel is so shaped that the weight of mercury forced over into the vessel *B*, which is fastened to one arm of a balance, is proportional to the square root of the differences in level. The pointer *P* then moves distances proportional to $\sqrt{p_1 - p_2}$. By ingenious modification of this idea, which is old, and additional adjustments to automatically allow also for density variations, Gehre has produced an apparently satisfactory commercial instrument. The flow may be read directly from an integrating attachment, which is of advantage, or it may be autographically recorded, or both methods may be used in the same instrument. The accuracy is guaranteed to within ± 5 per cent.

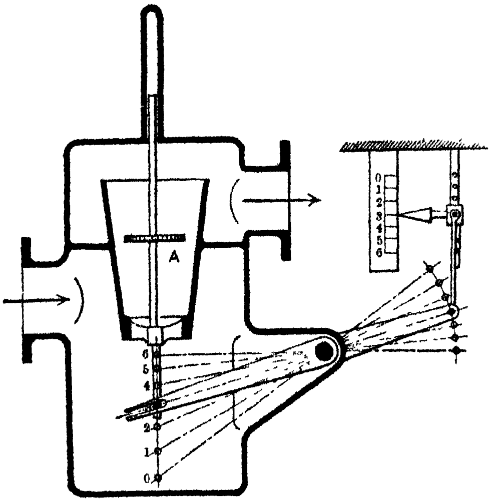

FIG. 342. — DIAGRAMMATIC SKETCH OF FLOAT STEAM METER.

(*d*) *Float Meters* are of two classes: those in which the float moves against a constant resistance but in which the free passage for steam increases with the magnitude of the movement and those in which the movement takes place against an increasing resistance but the free passage remains constant. Nearly all true float meters are of the first type.

The diagrammatic sketch, Fig. 342, shows the principle. The seat of the float *A* is extended in the direction of the current. If a simple conical form is used the effective cross section of free passage is only approximately proportional to the movement of the float, but the degree of approximation is sufficiently close for most purposes. The force required to lift the float, and consequently also the pressure difference (drop of pressure), remains constant. This pressure difference is kept so small that the effect on the density of the steam may be neglected and the formula for flow of liquids,

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* Gehre Dampfmesser Gesellschaft, Berlin. See the above-mentioned articles by Bendemann.

$$W = CF\sqrt{p_1 - p_2},$$

may be employed. Since $\sqrt{p_1 - p_2}$ is constant, but F varies in direct proportion to the lift of the float, the quantity W is practically a linear function of the lift.

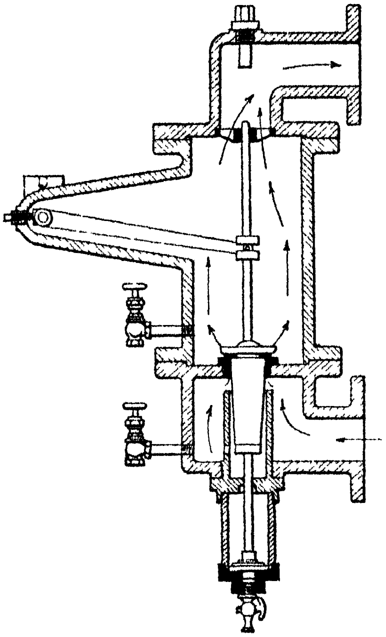


FIG. 343. — CROSS SECTION OF ST. JOHN STEAM METER.

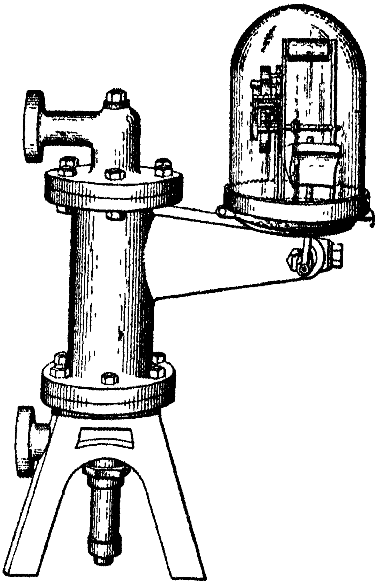
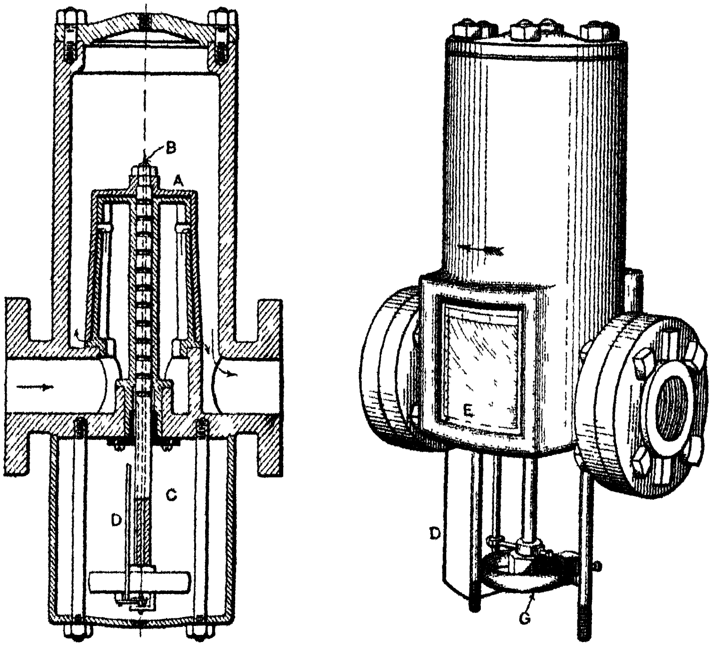


FIG. 344. — GENERAL VIEW, ST. JOHN STEAM METER.

This principle is in this country applied commercially in the St. John and in the Sargent steam meters. Fig. 343 shows a cross section of the former. The construction differs from that of Fig. 342 in that the float itself is made conical. This destroys the simple relation of proportionality between lift and quantity of flow, because the effective area of the float changes. Correction for this is apparently made in the construction of the registering apparatus which is connected as shown in Fig. 344. The record may be directly planimetered. Variation of density with pressure is not taken into account automatically by this meter.

The Sargent indicating steam meter is of later origin. It was first described by its inventor in the Transactions of the American Society of Mechanical Engineers, 1905, but since that time has apparently undergone considerable changes in construction. Figs. 345 and 346 are reproduced from Bendemann's discussion. The valve *A* is a conical float fitting over a conical seat and closing the passage completely with its lower edge when at rest. During operation the steam lifts the float proportional to the quantity passing. The valve spindle *B* is prolonged downward into the space *C*, which is open to the atmosphere. At the lower end the spindle carries a Bourdon tube, see *G* in Fig. 346, to which the pressure is transmitted through a hole in the spindle. The Bourdon tube carries a pointer *D* which moves



FIGS 345 AND 346.—SARGENT STEAM METER.

over a chart *E*, Fig. 346. The pointer has therefore a double motion, up or down, depending upon the lift of the valve, and in a horizontal plane, depending upon the steam pressure. The correction for density is therefore automatic.

As far as a comparison between orifice and float meters is concerned it might be said that they are probably not far apart on the score of accuracy. Where the steam main is inaccessible, as may sometimes be the case, the orifice meter is probably better, because its registering mechanism may be placed at some distance from the orifice and may be better taken care of.

IV. CALORIMETRIC MEASUREMENT OF VAPORS.

246. Methods Used. — The two calorimetric methods described in connection with the methods of measuring gases can be used for the measurement of vapors. The heat added either vaporizes entrained moisture or superheats the vapor. If the apparatus be so constructed that the vapor is slightly superheated before entrance, the determinations are generally fairly accurate.

One method of calorimetric determination which is applicable to vapors but not to gases is of interest. If vapor at a known temperature and pressure and with a known quality be condensed in a known weight of liquid, the weight of vapor condensed can be found from the temperature rise of the liquid. This method is used in connection with condensing calorimeters where the equations applying to the case may be looked up, see Chap. XIV.

247. The Flow of Steam in Pipes. — This problem is mentioned here merely with a view to indicating approximately what the flow of steam through any given pipe may be expected to be under given conditions.

An approximate formula may be derived by assuming that the velocity of the steam is not to exceed 6000 feet per minute. If we let W = the steam flowing in pounds per minute, δ the density of the steam in pounds per cubic foot, and d = diameter of the pipe in inches, we will have with the above assumption of velocity

$$W = 33\,\delta d^2 \text{ lbs. per min.} \qquad (67)$$

This formula is considered good for pipe sizes between 4 and 8 inches and for lengths under 100 feet. For smaller sizes than 4 inches the discharge will be less, and for the small sizes of pipe, especially if they are of considerable length, the formula is probably useless.

G. F. Gebhardt, in his "Steam Power Plant Engineering," has examined a number of formulæ established by various authorities for the flow of steam in pipes. He shows that in general they are all based upon the loss of head H expressed by*

$$H = f\,\frac{v^2}{2g}\cdot 4\frac{L}{d},$$

which is the same expression also used in the flow of water, see Art. 218.

For general practice, Gebhardt recommends the following formula:

$$W = 87\left\{\frac{\delta P d^5}{L\left(1+\frac{3.6}{d}\right)}\right\}^{\frac{1}{2}} \text{ lbs. per min.} \qquad (68)$$

in which

W = weight of steam in pounds per minute.
δ = mean density in pounds per cubic foot.
P = drop in pressure. Assumed at the figure desirable in any given case.
d = diameter of pipe in inches.
L = length of pipe in feet.

This strictly applies to well-lagged pipe only. As in the case of the flow of water the length L is not only the actual length, but the added friction of fittings should be taken care of by adding a certain equivalent length for each fitting. But few experiments have been made on the friction of fittings, and those give anything but concordant results. Until more satisfactory figures are available, those given by Briggs (Warming Buildings by Steam) and quoted by Gebhardt will have to serve.

For a standard 90-degree elbow or for a tee, the allowance should be about

$$L' = \frac{6.3\,d}{1+\frac{3.6}{d}} \text{ feet.}$$

The resistance offered by a gate valve when wide open may be neglected, while the allowance for a globe valve will be about $\frac{3}{2}$ times that for an elbow.

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* *Power*, June, 1907.

CHAPTER XIII.

COMBUSTION AND FUELS.

248. Combustion. — Combustion or burning is any kind of chemical combination evolving heat. The only kind of combustion used to produce heat for engineering purposes is the combination of various kinds of fuels with oxygen. In the ordinary sense the word *combustible* implies a capacity for combining rapidly with oxygen to produce heat.

249. Weight and Volume Relations. — The study of chemistry shows that when chemical reactions occur between elements or compounds there are involved definite weight relations characteristic of the particular elements acting and definite volume changes corresponding to the number of gaseous molecules reacting and produced. The reaction is a rearrangement of the atoms of the chemical elements into a new set of molecules. The volume of solids is in general negligible in comparison with that of gases of the same weight. For example, take the reaction

$$\begin{array}{rcl} C_2 + 2\ O_2 & = & 2\ CO_2 \\ 1\text{ lb. } C_2 + 2.664\text{ lbs. } O_2 & = & 3.664\text{ lbs. } CO_2 \\ 0.0071\text{ cu. ft. } C_2 + 29.85\text{ cu.ft. } O_2 & = & 29.85\text{ cu. ft. } CO_2. \\ \text{(solid)} \qquad\qquad \text{(gaseous)} & & \text{(gaseous)} \end{array}$$

The gas volumes above are under standard conditions, i.e. 32° F. and 14.7 pounds pressure per square inch.

Air is a mixture (not a chemical compound) of the gases nitrogen (N_2) and oxygen (O_2), and slight amounts of moisture (H_2O), carbon dioxide (CO_2), argon, and other inert gases. Considering the inert gases as nitrogen, the analysis of *dry air* is:

	By Volume	By Weight
N_2	79.00	76.74
CO_2	0.04	0.06
O_2	20.96	23.20

For engineering purposes the N_2 and CO_2 may be added together and considered as N_2. It is sometimes convenient to treat air as a gas with a molecular weight of 28.94 (based on $O_2 = 32.00$).

The following table gives weights and volumes of the principal substances concerned in the combustion of ordinary fuels:

Substance.	Chemical Symbol and Weight of		Weight in Pounds per Cu. Ft. at 32° F. and 14.7 Lbs. per Sq In. (One Atmosphere) Pressure (Standard Conditions).*
	Atom.	Molecule.	
Carbon	$C = 12.01$	...	145 (Solid)
Sulphur	$S = 32.07$	...	125 “
Oxygen	$O = 16\ 00$	$O_2 = 32.00$	0 08922 (Gas)
Nitrogen	$N = 14\ 04$ †	$N_2 = 28.08$ †	0 07830 “
Hydrogen	$H = 1\ 008$	$H_2 = 2\ 015$	0.00562 “
Marsh gas, or methane	...	$CH_4 = 16\ 03$	0.0447 “
Air	...	(28.94)	0 08071 “
Carbon dioxide	...	$CO_2 = 44\ 01$	0.12268 “
Carbon monoxide	...	$CO = 28.01$	0 07807 “
Water	...	$H_2O = 18.02$	(See steam tables) Vapor
Sulphur dioxide	...	$SO_2 = 64\ 07$	0.1786 (Gas)

* With sufficient accuracy the weight in pounds per cubic foot of a non associated dry gas, at standard conditions, is 0.002789 times the molecular weight of the gas.

† The "nitrogen" of the above table is "atmospheric" nitrogen, differing from chemically pure nitrogen in being mixed with a slight amount of the inert gas argon Chemical nitrogen has an atomic weight slightly under 14.01.

250. Heat of Combustion or Calorific Power. — The value of a fuel for heat production is the number of British thermal units of heat which may be generated by the complete combustion of one pound of the fuel, or in the metric system the number of large calories generated by one kilogram. This number of heat units, which is a constant for any given fuel irrespective of the manner in which the combustion is carried on, as long as it is complete, is known as the *heating value*, or *calorific power*, of that particular fuel. The relation of B.t.u. per pound to Kg-calories per kilogram is 9 : 5 or 1.8 : 1.

251. Combustion of Elementary Fuels. — The following table gives the heating values of the principal elements occurring in fuels, and data on the combustion of these elements:

Fuel of which Unit Weight is Burned.	Heating Value.		Oxygen, Required Weight.	Product of Combustion and Weight of Same.
	B.t.u./Lb.	Calories, Kilogram		
Hydrogen, H_2	61,950*	34,400	7.94	H_2O, 8.94
Carbon, burned to CO	4,380	2,430	1.332	CO, 2.332
Carbon, burned to CO_2	14,540	8,080	2.664	CO_2. 3.664
CO burned to CO_2	4,380	2,430	0.571	CO_2, 1.571
Sulphur, S, burned to SO_2	4,020	2,230	0.999	SO_2, 1.999
Sulphur, S, burned to SO_3	5,940	3,300	1.447	SO_3, 2.447
Iron, Fe, burned to FeO	2,430	1,350	0.286	FeO, 1.286

* Higher Heating Value.

252. Combustion of Hydrogen or of Fuels Containing Hydrogen. Higher and Lower Heating Value of Fuels. — When a fuel contains hydrogen a complication occurs, due to the fact that the water formed by the union of hydrogen and oxygen may or may not be condensed. If such water vapor is completely condensed, the heating value of the fuel is the so-called *higher heating value;* if the water vapor is not condensed, but passes off as steam, we obtain the *lower heating value.* The difference between the two heating values is the total heat of the steam as it escapes, less the sensible heat of the same weight of water at the temperature of the fuel and oxygen before combustion (room temperature). The total heat above 32° in one pound of steam, in the form of "humidity" in air or gases of combustion, is $(1058.7 + 0.455\ t_1)$ B.t.u.,* t_1 being the temperature of the air or gases of combustion. If room temperature be taken as t_2° F., the sensible heat of one pound of the steam condensed is $(t_2 - 32)$ B.t.u. One pound of steam escaping uncondensed in the products of combustion, at temperature t_1° F., will then take away $(1058.7 + 0.455\ t_1 - t_2 + 32.0)$ or $(1090.7 + 0.455\ t_1 - t_2)$ B.t.u. One pound of hydrogen burns to 8.94 pounds of water; hence per pound of hydrogen the loss is $8.94\ (1090.7 + 0.455\ t_1 - t_2)$ B.t.u. Assuming t_2, room temperature, at 60° F. (standard practice), the difference between the higher and the lower heating values of hydrogen shows the following values for various temperatures of escaping steam:

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* This equation is the result of plotting the total heat in superheated steam of low pressure (less than 3 lbs. absolute, as even that is beyond the partial absolute pressure of water vapor occurring in flue gases or exhaust gases) against temperature. The expression is therefore in a sense empirical, but holds good for flue or exhaust gas computations.

t_1° F., Temp of Escaping Steam.	Difference of Heating Values, B.t.u.	t_1° F., Temp. of Escaping Steam.	Difference of Heating Values, B.t.u.
60°	9,450	700°	12,060
300°	10,430	1000°	13,280
500°	11,240	1500°	15,320

The experimental higher heating value of hydrogen is 61,950 B.t.u.; if the products of combustion are brought back to 60° F., but uncondensed, the lower heating value is $61{,}950 - 9450 = 52{,}500$ B.t.u. The difference is 15 per cent of the higher, 18 per cent of the lower heating value.

So large a difference is not negligible; we must decide whether we are to charge a furnace with the higher or the lower heating value of a fuel containing hydrogen. The German custom is to use the lower heating value; the American tendency is to use the higher. No actual furnace or engine does cool the products of combustion so as to condense the steam in them. Calorimeters, which are used to find experimentally the heating value of fuels, generally condense only a part of the steam, and consequently give neither the higher nor the lower heating value directly. Theoretically perfect absorption of heat after combustion would condense all moisture formed from hydrogen. *It seems right to charge against a furnace or an engine what a theoretically perfect apparatus would do.*

The volume of the products of combustion may be different from the volume of the combustibles, or the gases entering the combustion may not be saturated with "humidity," so that a larger amount of moisture might remain at room temperature as "humidity" in the burned than in the unburned gases. The correction for humidity is very small if the air supplied to a furnace is saturated or nearly so; but in the case of ordinary coal burning, with the air supplied about 50 per cent saturated, over one-fourth of the moisture formed from hydrogen combustion would remain uncondensed if the products of combustion were cooled back to room temperature.

253. Temperatures Obtained in Combustion. — The calculation of the temperatures obtainable in combustion is made from the amount of available heat and the amount and specific heat of the products of combustion. Specific heats are not constant, but increase with temperature. The values in the last column of the table below, intended for use in ordinary combustion calculations, are nearly correct for a temperature of 600° F., a common temperature for flue gases.*

MEAN SPECIFIC HEATS AT CONSTANT PRESSURE, FOR COMBUSTION AT ATMOSPHERIC PRESSURE.

Substance.	Specific Heat	
	At Room Temperature 60° F.	At Ordinary Flue Gas Temperature 600° F.
Air	0.237	0.243
Oxygen	0.217	0.220
Nitrogen	0.243	0.249
Hydrogen	3.380	3.54
Carbon monoxide	0.243	0.249
Carbon dioxide	0.201	0.222
Steam, in form of "humidity"	0.455	0.462
Sulphur dioxide	0.154	0.17
Ash	0.2±	0.2±
Copper	0.093	
Iron	0.112	
Brass	0.093	
Glass	0.19	
Zinc	0.0935	
Wood	0.6	

The last six items are added to this table, as this data is sometimes required for calorimetric work.

To take an actual case, suppose we desire to compute the maximum possible temperature attainable by the combustion of carbon. Burning carbon in pure oxygen, the heat evolved from one pound of carbon is 14,600 B.t.u., and there are formed 3.664 pounds of CO_2. Taking the specific heat of CO_2 as 0.222 (it would really be higher at high temperatures), the temperature should be $14{,}600 \div (0.222 \times 3.664) = 18{,}000°$ F. Actually, however, the combustion cannot be made to go on rapidly enough to reach this theoretical temperature; heat will be lost too rapidly by radiation, and it is impossible to avoid heating surrounding materials as well as the products of combustion.

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* These values were obtained from comparison of numerous sources, greater credence being given to recent determinations at the Reichanstalt. For an extended discussion of the variations of specific heat see Chap. XXI.

If hydrogen be taken as the fuel, the available heat for raising the temperature of the products of combustion above 60° F. is the lower heating value for 60° F., 52,500 B.t.u. per pound. For combustion in oxygen the theoretical temperature is then $52{,}500 \div (0.462 \times 8.94) = 12{,}500°$ F. Despite this lower theoretical temperature, we actually obtain higher temperatures of combustion from hydrogen than from carbon, for in practice the problem becomes one of maximum energy development in a given space.

The principal reason for the low temperatures obtained in actual combustion is the dilution of the products of combustion with inert material. Every pound of O_2 in air carries with it $76.8/23.2 = 3.31$ pounds of N_2. Taking this into account for the combustion of carbon, using only the air necessary to supply the oxygen needed, we get $14{,}540 \div (0.222 \times 3.664 + 0.249 \times 2.664 \times 3.31) = 4900°$ F. If we used 1.5 times as much air as was needed to supply O_2 for the combustion of carbon, the theoretical rise of temperature during combustion would become 3360 degrees; with 2.0 times the needed air, 2550 degrees. By comparing these values with the 18,000 degrees for combustion in pure oxygen we see how large is the effect of dilution. Roughly, the theoretical temperature of combustion of carbon with air is 5000° F., divided by the "dilution coefficient," or ratio of air actually used to that necessary to supply oxygen to the carbon.

For some years past it was thought that the reason we cannot, in practice, realize the high theoretical combustion temperatures indicated above, was a dissociation of the CO_2 or H_2O; that is, the chemical compounds of C or H with O were supposed to break down spontaneously into their elements when the temperature became high. It is now known that dissociation of either CO_2 or H_2O occurs only to a very slight extent (less than 1 per cent) even at 3000° F. (1600° C.). Dissociation therefore plays little part in any commercial case of combustion.

254. Calorimetry. — A calorimeter, broadly speaking, is any device for measuring heat or heating value. A fuel calorimeter is a device in which a fuel sample may be completely burned, while the heat it produces is absorbed and measured. The combustion of the fuel must be complete. All the heat produced must be absorbed and measured. Heat from other sources should, as far as possible, be excluded, and allowance made for that which cannot be excluded. These requirements apply to all types of calorimeters.

In operation, calorimeters fall into two classes, the *discontinuous* and the *continuous types*. Solid fuels are nearly aways tested in calorimeters of the discontinuous type, burning a weighed sample in a calorimeter using a fixed amount of heat-absorbing body. Liquid fuels are occasionally tested in the same way. Liquid fuels are usually, and gaseous fuels practically always, tested in continuous or constant flow calorimeters, both fuel sample and heat-absorbing body being under flow at measured rates. It may be said that both theoretically and practically the continuous-flow calorimeter is superior to the discontinuous, because of greater ease of handling and freedom from extraneous heat losses of the " radiation type."

The *discontinuous calorimeters* will be discussed first. The parts of such a calorimeter are in general as follows:

(1) Crucible for holding sample of fuel.

(2) Igniter for firing the sample.

(3) Combustion chamber.

(4) Apparatus for supplying oxygen for combustion, for only in oxygen can complete combustion be assured in these calorimeters.

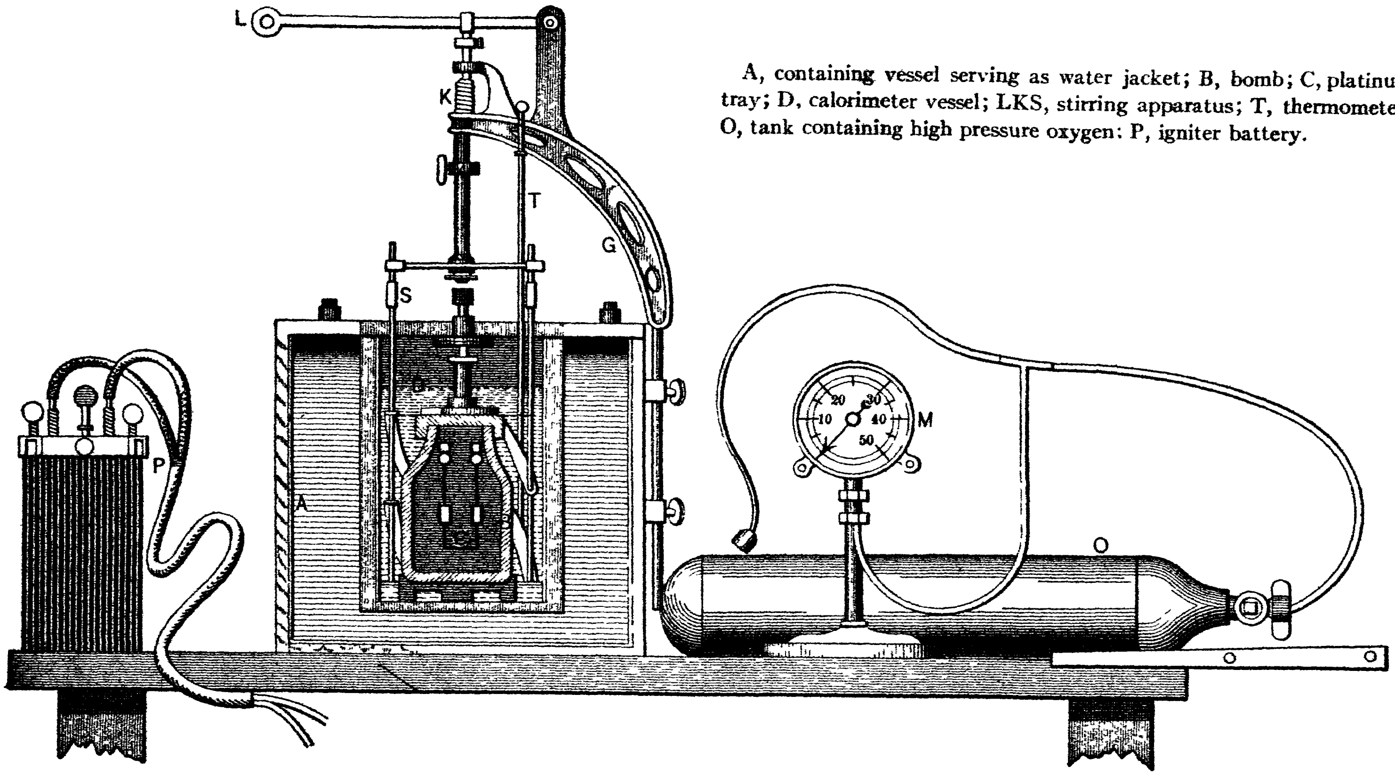
(5) Heat-absorbing body surrounding the combustion chamber.

(6) Thermometer or other device for measuring temperature changes in the heat-absorbing body, and thus measuring the heat absorbed.

(7) Jacket for excluding extraneous heat.

The general problems of discontinuous calorimetry will be discussed under the description of the *bomb calorimeter*, which is the most accurate, reliable, and also the most expensive of the calorimeters for solid fuels.

255. The Bomb Calorimeter. — There are a number of forms of this calorimeter now on the market, nearly all of them modifications of the original Berthelot type. The essential parts in every case are: (*a*) a strong steel vessel, usually lined with some non-corroding substance like platinum, gold, porcelain, or nickel, serving as a combustion chamber; (*b*) a small crucible, usually of platinum, inside of the vessel, to hold the sample; (*c*) valves or connections for



A, containing vessel serving as water jacket; B, bomb; C, platinum tray; D, calorimeter vessel; LKS, stirring apparatus; T, thermometer; O, tank containing high pressure oxygen: P, igniter battery.

FIG 347.—MAHLER FUEL CALORIMETER.

charging the bomb with oxygen; (*d*) means for igniting the fuel; (*e*) a calorimeter vessel filled with water into which the bomb is placed; (*f*) a second vessel to serve as a jacket for the calorimeter vessel to reduce heat losses; (*g*) thermometers to measure temperature; and (*h*) stirring apparatus to maintain uniform temperature in the water. These parts will be recognized in the following cuts, in which Fig. 347 represents the Mahler, Fig. 348 the Hempel, Fig. 349 the Atwater, and Fig. 350 the Emerson calorimeter.

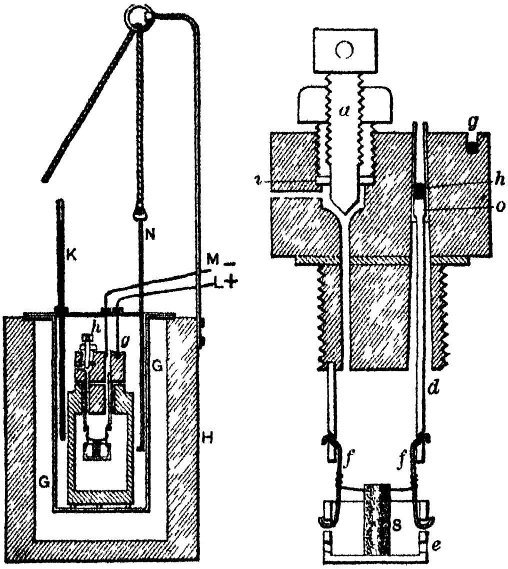


FIG. 348.—HEMPEL'S FUEL CALORIMETER WITH ENLARGED CHARGING PLUG

The "bomb" is a heavy-walled combustion chamber in which the oxygen for the combustion is held under a pressure of 125 to 300 pounds to the square inch. By using the high pressure, a considerable amount of gaseous oxygen can be put into a small space, surrounding the fuel to be burned; more oxygen, of course, being put in than is needed for complete combustion of the fuel.

The oxygen may be obtained either by chemical reaction between certain substances (as, for instance, manganese dioxide and chlorate of potash, equal parts) or by electrolysis, or by purchasing it in steel vessels compressed to about 20 atmospheres. In the latter shape it may be obtained from any wholesale chemical house, which is perhaps the simplest and best way of obtaining this necessary gas. The method first mentioned may be carried out as shown in Fig 351, in which the crucible *C* contains the mixture of chemicals The latter is heated and the oxygen gas generated flows through *dcb* into the calorimeter. A gauge *B* indicates the pressure attained. The vessel *E* is filled with water to cool the gas. Any chlorine gas coming over may be removed by first passing the gas through a close roll of brass-wire gauze.

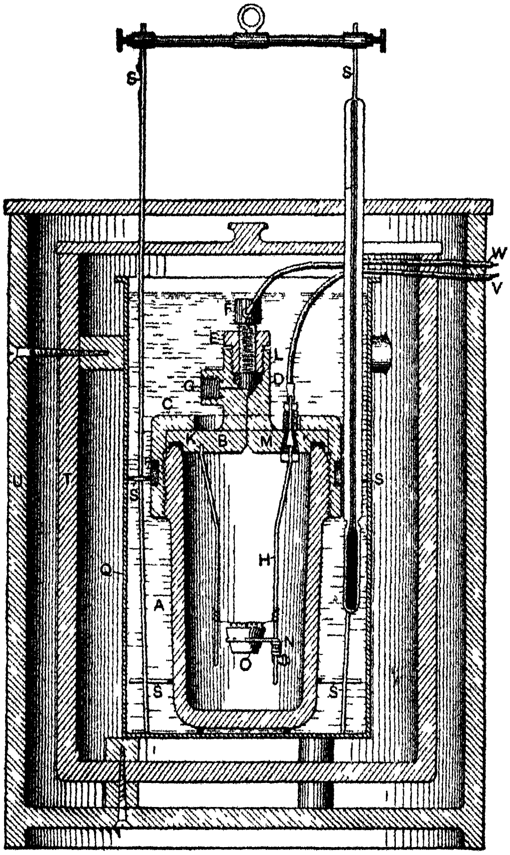


Fig 349 —Atwater Bomb Calorimeter

Ignition of the fuel is generally electrical, and is accomplished either by fusing and burning a piece of fine iron wire in contact with the fuel, or by heating to redness a platinum wire dipping into the fuel. Heat from the igniting device is added to that given out in the combustion of the fuel, and correction must be made for this extra heat from the igniter. This igniter heat may easily amount to 0.3 to 0.5 of one per cent of the heat developed in the calorimeter.

The bomb is immersed in a small body of liquid, usually water.

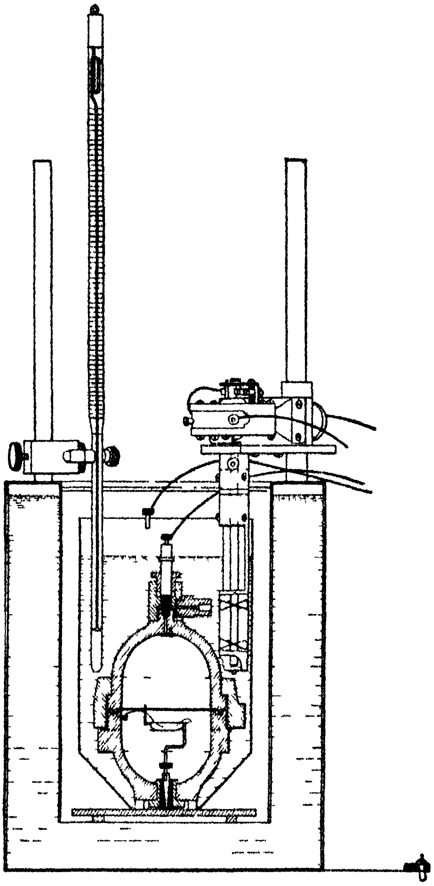


FIG 350 — EMERSON BOMB CALORIMETER

The heat from the combustion raises the temperature of the bomb and of this liquid and its containing vessel. This rise of temperature of the bomb, water, and containing vessel, when corrected for radiation and similar heat exchanges, measures the heat evolved in the bomb In practice the absorption of heat by the bomb and the containing vessel is equivalent to having a certain extra amount of water present to absorb heat; hence the term *water equivalent* for the summation of the products mass times specific heat for each of the metallic parts of the calorimeter which run through the same temperature changes as the water. In the most accurate work it should not be forgotten that the specific heats of both the water and the metallic parts change with temperature. The magnitude of this change is not likely to exceed 0.3 of one per cent. It is uniformly neglected in commercial work.

The temperature of the water, bomb, and containing vessel is measured by a mercury thermometer or some equivalent device, such as a spiral wire of which the electrical resistance is measured. To make sure that the temperature reading (usually a measurement taken at one spot only) represents the temperature of the entire instrument, stirring of the water is necessary.

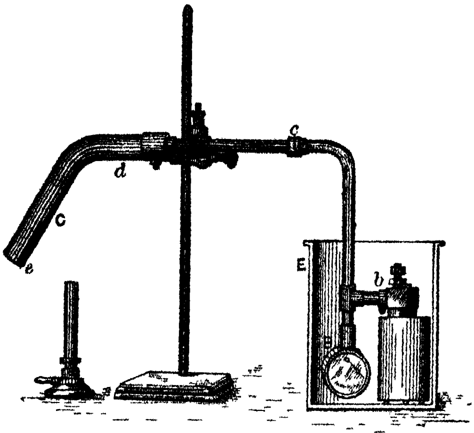


Fig 351 —Apparatus for Preparing Oxygen.

If the instrument could be thermally isolated from its surroundings, it would be necessary only to measure the temperature before ignition and the temperature after complete combustion, calculating the heat evolved from the temperature range and mass and specific heats of the calorimeter parts absorbing heat. Such thermal isolation is impossible; the calorimeter is continually exchanging heat with its surroundings by (1) direct radiation, (2) convection currents of air, (3) evaporation of water from the calorimeter to the atmosphere, which is practically never saturated with moisture.

To reduce these heat exchanges — usually classed as "radiation," although real radiation loss is the smallest of the exchanges mentioned — to their lowest terms, it is customary to surround the containing vessel of the calorimeter with a "jacket" and to cover the calorimeter so as to prevent access of air and consequent convection currents. The containing vessel of the calorimeter is made of polished metal to diminish radiation. The jacket is a double-walled vessel, polished at least on the inside, with the space between its walls filled with water or other liquid, so as to keep the jacket fairly constant in temperature.

It is usually assumed that for a jacketed calorimeter the "radiation rate," or rate of loss or gain of heat by the calorimeter in exchange with its surroundings, is proportional to the difference of temperature between calorimeter and jacket. Analysis shows that this assumption is not justified. True radiation between calorimeter and jacket is closely enough a straight-line function of the temperature difference between them. Convection transfer of heat between calorimeter and jacket is also probably given nearly enough as a straight-line function of the same temperature difference. These heat exchanges reverse in direction and amount when the temperature difference between calorimeter and jacket changes sign. But there is also a slight amount of convection loss of heat from calorimeter to the outer atmosphere, or room, when the calorimeter is hotter than the room — a heat exchange outside of the control of the jacket. This exchange is not reversed when the calorimeter is colder than the room, for the space between jacket and calorimeter then fills with dead air, colder than the room, and the exchange ceases. Frequently the most important of all these extraneous heat effects, on the calorimeter, is the heat loss by evaporation of water into the unsaturated atmosphere of the room. This heat loss depends on the temperature of the calorimeter, increasing in rate as the temperature of the calorimeter rises, and on the relative "humidity" of the air in the room. This effect is always a loss and is independent of the jacket.

Summing up all of these heat exchanges between the calorimeter and its surroundings, it will be seen that the "radiation rate" is dependent on so many different things that it must, for accuracy, be determined for the beginning and end of each run of the calorimeter. The best simple assumption as to variation of the "radiation rate" between the two end conditions of the run is that *the radiation rate varies directly with the temperature of the calorimeter.* This is better than to assume straight-line variation with temperature difference between calorimeter and jacket, because this temperature difference controls only part of the various heat exchanges, while the calorimeter temperature enters as a controlling factor into all. The common assumption that "radiation" is zero or negligible, when the initial calorimeter temperature is just as much below the jacket temperature (or room temperature) as the final calorimeter temperature is above the same, is quite unjustified.

256. Method of Operating a Bomb Calorimeter. — The conduct of a determination may now be considered. The calorimeter and accessory apparatus are assumed to be clean and in working order. The first thing to be done is to find the "water equivalent" of the calorimeter, unless this is already known. This may be done in one of three ways:

(*a*) *Calculation.* — Weigh separately each of those parts of the instrument that go through the same temperature range as the water. Multiply each weight by the specific heat of the material of which the part is made. The summation of these products is the water equivalent. Allowance must be made for the poor conductivity of wood or glass, which generally acts so that only a surface layer of such material is thermally active during the running of the calorimeter; in a less degree such an allowance should be made for neutral parts projecting markedly above the surface of the water.

(*b*) *Method of Mixtures.* — Partly fill the calorimeter vessel with a known mass of water. Allow all parts to come to thermal equilibrium, then determine temperature and radiation rate. Add a known mass of water (completing the filling of the calorimeter), of a known temperature considerably higher or lower than that of the water already in the calorimeter. Stir to equilibrium, taking readings of temperature, and finish with a determination of the final radiation rate. After correction for radiation the discrepancy in the heat balance is the product of the water equivalent and the observed temperature range. Make several determinations and average the results.

(*c*) *Method of Adding a Known Amount of Heat Inside the Bomb.* — This may be done by (1) burning a sample of fuel the value of which is already well known; (2) inserting an electrical resistance heater inside the bomb and measuring the input to the heater with an accurate wattmeter. Runs are made as with an unknown fuel, and the calculations worked backward to find the water equivalent. Methods (*b*) and (*c*) are to be preferred to method (*a*) for accuracy. The water equivalent of a calorimeter is usually determined once for all. There is no reason why it should change with time, but it does change slightly with temperature, because of change of the specific heat of the parts.

Knowing the water equivalent, the regular run proceeds as follows: Weigh out into the crucible of the calorimeter a proper amount of sample, usually from half a gram to two grams, depending on the make and size of the calorimeter. The sample, if of coal, should be in a finely divided condition, about fine enough to pass a 100-mesh sieve. It should preferably be somewhat moist, as moderately moist coal burns less explosively than dry coal, particularly if the sample is of the bituminous or lignite variety. Hence the calorimeter sample may best be made up from the coal " as received." (See later directions for coal analysis.) Put the crucible with the weighed sample into the bomb and arrange the igniter for operation. Put a few drops of water into the bomb. This insures an initially saturated atmosphere in the bomb, and hence the finding of " higher heating value " from any hydrogen. Screw the top onto the bomb and charge the bomb with oxygen gas to the correct pressure. Close the charging cock, and put the charged bomb into position in the calorimeter vessel. Fill the calorimeter vessel with a weighed amount of water. Weighing is more accurate than measuring. Arrange stirrers, thermometers, etc., ready for operation. The calorimeter liquid should not be over a degree or two colder or hotter than the jacket, and neither ought to be more than five degrees different from room temperature, although this is less important. Let the calorimeter stand a few minutes, watching the temperature and occasionally stirring, until the parts are in a settled condition. Then determine the initial radiation rate by temperature observations at one-minute intervals extending over five or more minutes, stirring between readings. When the readings have shown a satisfactory constant rate of change of temperature (so-called " radiation rate "), start the igniter on some even minute. Thereafter take readings every half minute until combustion is over and the calorimeter settles to a second radiation period. Stir thoroughly between readings. Finish with a determination of final radiation rate similar to the initial. Remove the bomb from the calorimeter. Open the charging cock carefully to let out the gases. Then unfasten the cover and inspect the crucible and the interior of the bomb to make sure of complete combustion. Clean up for the next run.

The observations taken during a commercial test, and the method of working them up, are as follows:

SAMPLE CALCULATIONS FOR BOMB CALORIMETER.

Weight of coal sample, grams	1.2794
Weight of iron igniter wire burned, grams	0.0022
Weight of calorimeter water, grams	1408
Water equivalent of calorimeter, grams	345
Water + water equivalent, grams	1753
Pressure of oxygen charge in bomb	250 lbs. per square inch.

Observations of Run				Calculated Values (see below)					
	Time.	Observed Temp. in Calorimeter.		Mean Temp. of Interval	Radiation Rate for the Interval	Radiation for the Interval	Total Radiation Correction	Corrected Temperature	
		C°.							
	12 : 12	23.20	Initial radiation rate.				+0.1222	23.322	23.318
	14	23.22		23.21	−0.0137	−0.0274	+0.0948	23.315	
	16	23.25		23.23	−0.0136	−0.0272	+0.0676	23.318	
	18	23.28		23.26	−0.0136	−0.0272	+0.0404	23.320	
	20	23.30		23.29	−0.0135	−0.0270	+0.0134	23.313	
Run	21	23.32	Ignition.	23.31	−0.0134	−0.0134	0.0000	23.320	
	22	27.20		25.26	−0.0079	−0.0079	−0.0079	27.192	
	23	28.45		27.92	−0.0004	−0.0004	−0.0083	28.442	
	24	28.68		28.57	+0.0015	+0.0015	−0.0068	28.673	
	25	28.74		28.71	+0.0019	+0.0019	−0.0049	28.735	
	26	28.77		28.76	+0.0021	+0.0021	−0.0028	28.767	
	27	28.78+		28.78	+0.0021	+0.0021	−0.0007	28.784	
	28	28.79		28.79	+0.0021	+0.0021	+0.0014	28.791	
	29	28.79		28.79	+0.0021	+0.0021	+0.0035	28.794	
	30	28.79		28.79	+0.0021	+0.0021	+0.0056	28.796	
	32	28.79		28.79	+0.0021	+0.0042	+0.0098	28.800	
	34	28.79	Final radiation rate.	28.79	+0.0021	+0.0042	+0.0140	28.804	28.803
	36	28.78+		28.79	+0.0021	+0.0042	+0.0182	28.803	
	38	28.78		28.78	+0.0021	+0.0042	+0.0224	28.802	
	40	28.77+		28.78	+0.0021	+0.0042	+0.0266	28.802	
	42	28.77		28.77	+0.0021	+0.0042	+0.0308	28.801	
	44	28.77		28.77	+0.0021	+0.0042	+0.0350	28.805	

The first calculation is for the "radiation correction." The readings of the initial and final radiation-rate determinations should be plotted to an open

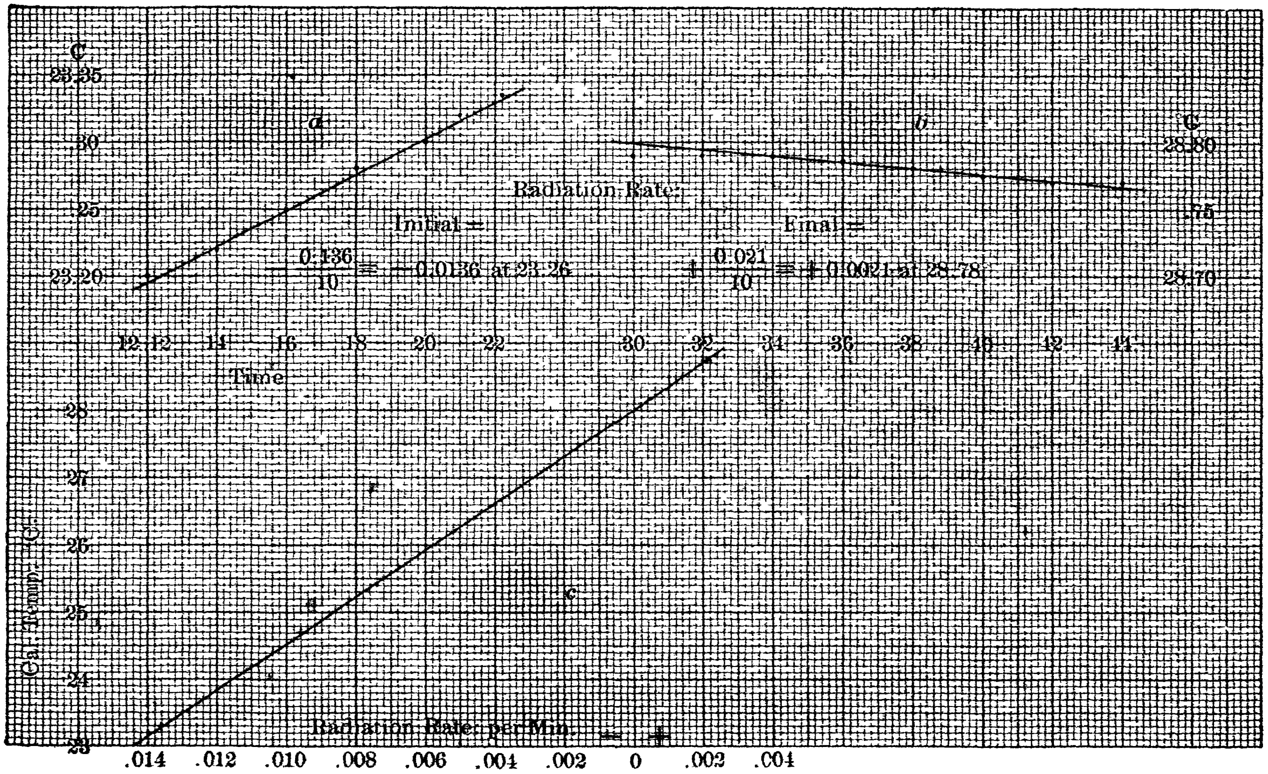

C
23.35
.30
.25
23.20
a
b
C
28.80
.75
28.70
Radiation Rate:
Initial =
−0.136/10 = −0.0136 at 23.26
Final =
+0.021/10 = +0.0021 at 28.78
12.12
14
16
18
20
22
30
32
34
36
38
40
42
44
Time
28
27
26
25
24
23
c
Cal. Temp. °C.
Radiation Rate per Min. − +
.014
.012
.010
.008
.006
.004
.002
0
.002
.004

FIG. 352.

temperature scale, as in Figs. 352a and 352b, and the slopes of the straight lines accurately determined, with the mean temperatures at which each rate is found. Then plot Fig. 352c, radiation rate against temperature in calorimeter, drawing a straight line through the two points determined.

1. *Long Method for Radiation Correction.* — Average the readings of the test in pairs successively, to give the column "mean temperature of interval." For each mean temperature of interval pick off from the curve, Fig. 352c, the "radiation rate for the interval." Multiply the radiation rate by the time length of the interval in minutes, getting the "radiation for the interval." Assume zero radiation correction at any arbitrary reading (in this case the reading at ignition). Then the "total radiation correction" at any other reading of the test is the summation of the values of "radiation for the interval" between this reading and the reading at which the radiation is assumed zero. The continued summation of the column "radiation for the interval" gives the column "total radiation correction." Total radiation correction added to the observed temperatures gives "corrected temperatures."

If the radiation correction has been properly made, the corrected temperatures for the radiation runs at the beginning and end of the test should be constant within the error of reading. This is very nearly realized, the average being 23.318° for the initial and 28.803° for the final radiation runs. Inspection of the column of "corrected temperatures," in comparison with that of "observed temperatures," will show that the observed temperatures reached a constant value for some time before the corrected temperatures did. This means that there was still going on in the calorimeter a slight evolution of heat which was practically balanced by radiation loss. *The run proper on the calorimeter must not be considered complete until a constant rate of change of calorimeter temperatures against time has been established.*

The corrected rise of temperature in the example here given is the difference of the average corrected temperatures for the final and initial radiation periods, or 28.803 − 23.318 = 5.485° C.

2. *Shorter Method for Radiation Correction.* — Plot Fig. 352a and Fig. 352b as before. Find, by the trapezoidal or Durand's rule, the time average temperature of the run proper, that is, from 12:21 to 12:34 inclusive in the example. The average temperature is 28.42° C. In Fig 352c find that the radiation rate corresponding to this average temperature is +0.0011 degree per minute. The time is 13 minutes. Hence, $13 \times 0.0011 = +0.0143$, the radiation correction for the run. The corrected temperature rise is then $28.79 + 0.014 - 23.32 = 5.484°$. This method of calculation is exactly equivalent to the long method above, but avoids practically all of the work. If, however, one took the run as lasting from 12:21 to 12:28, *as might be done, judging from observed temperatures only*, this short method would not show that the chosen length of run was in error, while the longer method does reveal such errors.

3. *Approximate Short Method for Radiation Correction.* — This assumes that the initial radiation rate applies until the time when a temperature is reached

midway between the initial and final temperatures, and after that time the final radiation rate applies. In the example given here, the average of initial and final temperatures is (28.79 + 23.32) ÷ 2 = 26.06. This temperature is reached in three-quarters of a minute from the start, as found by plotting a rough curve of temperature against time for the run. Hence we have:

Initial radiation rate for $\frac{3}{4}$ min., $\frac{3}{4} \times (-.0136) = -.0102$.
Final radiation rate for $12\frac{1}{4}$ min., $12\frac{1}{4} \times (+.0021) = +.0257$.
Total radiation correction, $+.0257 - .0102 = +.0155$.
Corrected rise, $28.79 + .015 - 23.32 = 5.485°$.

This result agrees exactly with that of Method 1, but this agreement is only accidental.

4. The method of calculating radiation correction for the run by applying the *algebraic mean* of the initial and final radiation rates to the entire time is incorrect; a little thought shows it to be irrational. It seems, however, to be quite commonly used.

5. *Calculations of Heat Evolved.* — The heat, in calories, evolved in the calorimeter is the product of the water + water equivalent in grams, multiplied by the corrected temperature rise in degrees C. In the example it is 1753 × 5.485 = 9615 calories. The combustion of the iron wire in igniting gave 1350 × .0022 = 3.0 calories. As much or more again was due to the electric current used to heat up the iron wire. Hence, we may say that 9615 − 6 = 9609 calories came from the coal burned. Since B.t.u. per pound = 1.8 times calories per gram, and weight of coal was 1.2794 grams, the heating power of the coal was

$$\frac{9609 \times 1.8}{1.2794} = 13,820 \text{ B.t.u. per pound.}$$

This calculation assumes the specific heat of water = 1.0000; the real specific heat for the mean temperature of the run is 0.9972. Remembering that the correction for this specific heat applies only to the 1408 grams of water out of the 1753 of (water + water equivalent), the corrected and final value for the heating power is 13,820 − 30 = 13,780 B.t.u. per pound.

In commercial work the corrections for heat from the igniter wire and for variation of the specific heat of water from unity are not ordinarily made.

Other Forms of Fuel Calorimeters.

257. Favre and Silbermann Fuel Calorimeter. — This apparatus, as shown in Fig. 353, consisted of a combustion chamber *A* formed of thin copper, gilt internally, and fitted with a cover through which solid combustibles could be introduced into the cage *C*. The cover was traversed by a tube *E*, connected by means of a suitable pipe to a reservoir of the gas to be used in combustion, and by a second tube *D*, the lower end of which was closed with alum and glass, transparent but adiathermic substances which permitted a view of the process of combustion without any loss of heat.

For convenience of observation a small inclined mirror was placed above the peep-tube *D*. The products of combustion were carried off by a pipe *F*, the lower portion of which constituted a thin copper coil, and the upper part was connected to the apparatus in which the non-condensible products were collected and examined. The whole of this portion of the calorimeter was plunged into a thin copper vessel, *G*, silvered internally and filled with water, which was kept thoroughly mixed by means of agitators, *H*. The second vessel stood on wooden blocks inside a third one, *I*, the sides and bottom of which were covered with swanskins with the down on, and the whole was immersed in a fourth vessel, *J*, filled with water

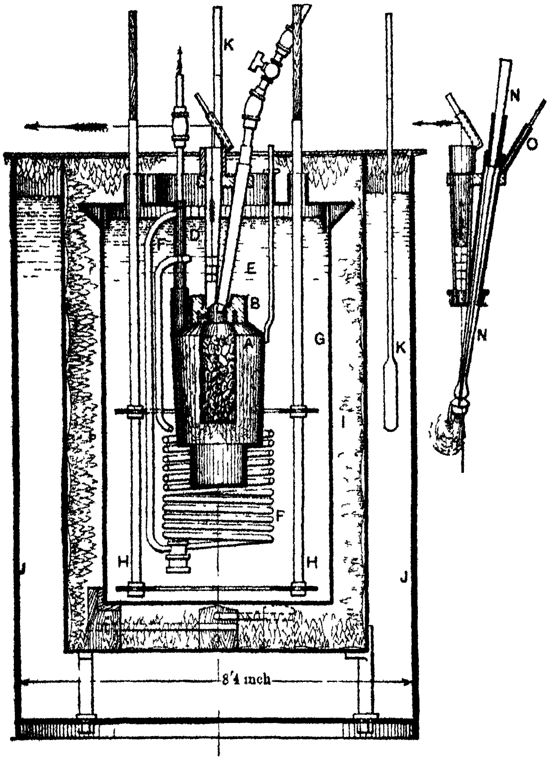


FIG. 353 —FAVRE AND SILBERMANN FUEL CALORIMETER

kept at the average temperature of the laboratory. Thermometers *KK* of great delicacy were used to measure the increase of temperature in the water surrounding the combustion chamber. The quantity of heat developed by the combustion of a known weight of fuel was determined by the increase of temperature of the water contained in the vessel *G*. For finding the calorific value of gases only, the cage *C* was removed and a compound jet, *NO*, substituted for the single gas pipe, ignition being produced by an electric spark or by some spongy platinum fixed at the end of the jet.

Combustion is slower than in the bomb, and radiation thus becomes more important, and introduces larger uncertainties into the results. Favre and Silbermann calorimeters will not be found in commercial work.

258. Carpenter Calorimeter. — The Carpenter calorimeter, Figs. 354 and 355, introduces a novelty in that it makes the calorimeter liquid do its own thermometry. For this purpose the calorimeter vessel is closed at the top, and is completely filled by the calorimeter liquid (usually water). The calorimeter vessel thus becomes practically the bulb of a huge thermometer.

The general appearance of the instrument is shown in Fig. 354; a sectional view of the interior part is shown in Fig. 355, from which it is seen that in principle the instrument is a large thermometer, in the bulb of which combustion takes place, the heat being absorbed by the liquid which is within the bulb. The rise in temperature is denoted by the height to which a column of liquid rises in the attached glass tube.

In construction, Fig. 355, the instrument consists of a chamber, 15, which has a removable bottom, shown in section. The chamber is supplied with oxygen for combustion through tube 24, 25, the products of combustion being discharged through a spiral tube 28, 29, 30.

Surrounding the combustion chamber is a larger closed chamber, 1, filled with water and connecting with an open glass tube 9 and 10. Above the water chamber 1 is a diaphragm 12, which can be changed in position by screw 14 so as to adjust the zero level in the open glass tube at any desired point. A glass for observing the process of combustion is inserted at 33, in the top of the combustion chamber, and also at 34, in the top of the water chamber, and at 36 in the top of the outer case

This instrument readily slips into an outside case, which is nickel-plated and polished on the inside, so as to reduce radiation as much as possible The instrument is supported on strips of felting, 5 and 6, Fig 355 A funnel for filling is provided at 37, which can also be used for emptying if desired

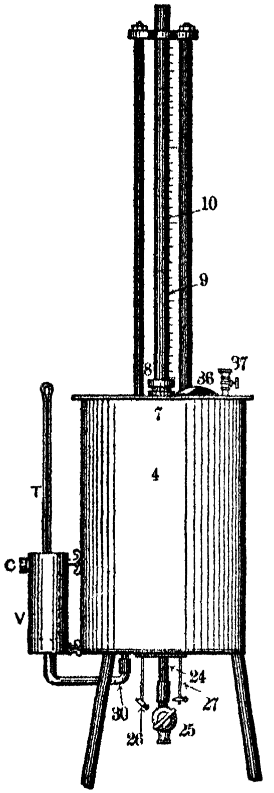


Fig 354

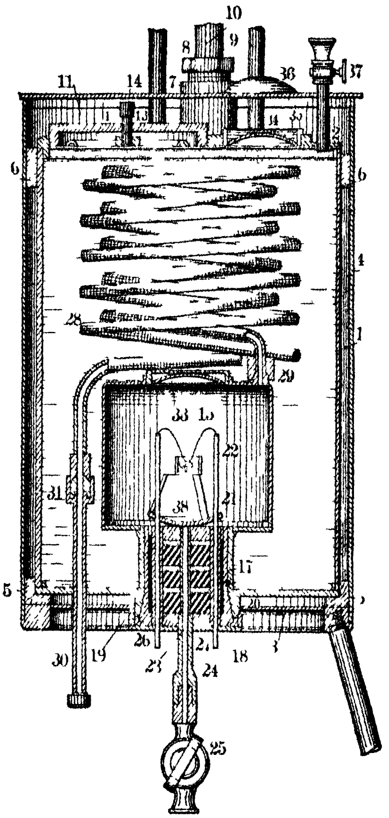


Fig 355

Carpenter Fuel Calorimeter

The plug which stops up the bottom of the combustion chamber carries a dish, 22, into which the fuel for combustion is placed, also two wires passing through tubes of vulcanized fiber, which are adjustable in a vertical direction, and connected with a thin platinum wire at the ends. These wires are connected to a source of electric current and used for firing the fuel. On the top part of the plug is placed a silver mirror, 38, to deflect any radiant heat. Through the center of this plug passes a tube, 25, through which the oxygen passes to supply combustion. The plug is made with alternate layers of rubber and asbestos fiber, the outside only being of metal, which, being in contact with the wall of the water chamber, can transfer little or no heat to the outside.

The discharge gases pass through a long coil of copper pipe, and are discharged through a very fine orifice in a cap at 30 in Fig. 355, or at *C* in Fig. 354.

The effect of the combustion is to expand the water and thus raise the outer level in the tube 10. The theory is that this expansion is a direct function of the amount of heat added to the water. In practice the rise in the tube must be corrected not only for radiation but also for the fact that the coefficient of expansion of water varies rapidly with the temperature. In the directions furnished with the earlier instruments the latter fact was not taken into account and radiation was accounted for simply by letting the calorimeter stand, after the combustion was completed, for the same length of time as was taken up by the combustion, noting the drop in the water level in the tube. This drop was then added to the rise obtained during combustion, on the assumption that radiation kept the rise down by the amount noted. This method is not rational.

It will be appreciated that the heat interchanges controlling the rise in the tube are rather complex, and the best way to obtain a standardization of the instrument is to calibrate by means of burning a certain quantity of a fuel of known heating value. The "calibration constant" will be a function of the temperature of the calorimeter liquid (on account of the change of coefficient of expansion with temperature), and a thermometer should therefore be provided. In the commercial form of this calorimeter as described this may be done by inserting a thermometer cup in place of the filler plug or funnel 37.

Adjustment of the height of liquid in the expansion tube is sometimes necessary. To make this adjustment, cut in between the stuffing box at the top of the calorimeter and the expansion tube a glass tee with a ground cock in the side branch, making connection by a rubber tube to a reservoir hung behind the scale and expansion tube. Adjustment of scale reading is made by opening the glass cock, moving the reservoir up or down, and closing the cock. When the calorimeter is not in use the cock should be open, so that as the calorimeter cools off the liquid may not contract so far as to draw air into the top of the calorimeter.

No stirring is done. The argument is that if one part of the liquid becomes hotter than the rest, it expands correspondingly, and hence changes the scale reading. As the scale shows the total expansion of the entire body of liquid, it indicates accurately the *mean* temperature of the liquid. Temperature differences in different parts of the liquid cannot become large, for convection currents would then be set up which would tend to restore equilibrium.

To make a run with the Carpenter calorimeter proceed as follows: See that the calorimeter is correctly set up inside its jacket, that thermometer and plug are in place, etc. Adjust the water level in the scaled expansion tube to be just above the zero of the scale. Weigh out a fuel sample (about one gram), preparing the sample as in the directions above for the bomb, by grinding some of the "coal as received" to about the fineness of a 100-mesh sieve. Remove the charging plug from the calorimeter, adjust the loaded crucible upon the stand, and set the igniter ready for operation. Screw the plug back into place. Connect up the igniter wires and the oxygen tube. Turn on the oxygen, adjusting the supply until it flows through under an entering pressure of about 3 inches of water, as indicated by a manometer connected on the line. Let the arrangement stand with oxygen on for about five minutes to become steady. Then take a reading of the scale and the thermometer. Five minutes later repeat these readings. These establish the initial "radiation rate." At the instant of this second set of readings operate the igniter to start the combustion, watching through the windows provided for that purpose. If the coal flames up considerably, choke the oxygen supply until the flame quiets. Take readings of the scale at least every five minutes, exactly on the minute, dating from the time of ignition. It is better to take readings rather more frequently, say every two or three minutes, while the combustion goes on. Combustion continues from ten to fifteen minutes. After combustion ceases it takes from ten to fifteen minutes longer for the calorimeter to steady on to the second radiation rate. Continue readings at five-minute intervals until a maintained steady change from reading to reading shows that the final radiation rate has been found. With the last scale readings observe also the temperature by the thermometer of the liquid in the calorimeter. Shut off the oxygen; remove the plug. Weigh the crucible and ash, *after drying*, to determine the ash content of the coal. The combustion in this calorimeter is so quiet that the calorimeter sample serves for the ash determination.

259. **Calculations with the Carpenter Calorimeter.** — The calculations and observations for this test are like those of the bomb calorimeter, with the substitution of calorimeter scale reading for calorimeter temperature. Radiation rates, rise, corrected readings, etc., are figured on the scale readings. Then, adding the actual observed temperatures of calorimeter liquid at beginning and end of run, and dividing by 2, find the actual mean temperature of the run. With this enter the calibration curve (of calorimeter constant against mean temperature of run), and find the particular value of the constant applying to the run. The calorimeter constant times the corrected scale rise gives the B.t.u. (or calories) produced by the burning of the sample. From there on the calculations are again like those of the bomb calorimeter.

It has been mentioned that calibration of this calorimeter must be made. A "known" fuel may be prepared very easily by either burning, coking, and grinding granulated sugar or by coking and grinding any good grade of soft coal. The coking should be done at white heat under exclusion of air. The coke thus made should be ground to pass a 50-mesh sieve, and should be used *dry* — that is, just previous to use it should have been dried by heating for an hour or more at 250° F. or higher. The combustible portion of the material so made may safely be assumed to be pure carbon, of a heating power of 14,540 B.t.u. per pound, equivalent to 32.20 B.t.u. per gram. The difference in weight of crucible + sample before the run and *dried* crucible + ash after the run gives the weight of carbon burned. For calibration, runs should be made with this known fuel exactly as with an unknown; then by calculating backwards the calorimeter constant for the mean temperature of each run is found. A calibration curve of calorimeter constant against mean temperature of run should then be plotted.

It will be noted that nothing has been said about the heat from the igniter. This should be controlled so as to be the same from run to run. Further, such a weight of sample should be taken as to make the total heat evolved in the calorimeter practically the same in every run. When these conditions are met no attention need be paid to the heat from the igniter, as it will form the same proportionate part of runs upon both known and unknown fuels; and the "constants" found in the runs on known fuels will apply to runs upon unknown fuels. If these conditions are not met the calorimeter will not give satisfactory results.

The actual run during which calorimeter readings are being taken lasts about half an hour with the Carpenter calorimeter. On account of this long time radiation mounts up, so that the radiation correction may become as much as 10 per cent of the total corrected scale reading. As the certainty of radiation calculations is of the order of ± 10 per cent, this introduces a possible error of ± 1 per cent, and better work than this should not be attempted with the Carpenter calorimeter. This accuracy is, however, good enough for most engineering work.

260. The Parr Calorimeter. — The Parr calorimeter, Fig. 356, in its general operation, method of run, etc., is essentially similar to the bomb calorimeters. It differs from them in that, instead of using compressed gaseous oxygen surrounding the fuel, it gets oxygen from a chemical powder (sodium peroxide) mixed with the powdered fuel. The reactions in this case, however, are not those of simple combustion. The products of combustion, CO_2, H_2O, SO_2, or SO_3, etc., further react with some of the Na_2O_2 or Na_2O to form Na_2CO_3, $NaOH$, Na_2SO_3, or Na_2SO_4, etc., and these products are therefore chemically bound and do not escape. These by-reactions also evolve heat, the quantity of which depends upon the proportion of the mixture used. In a way this is an advantage, in that it increases the amount of heat to be measured in the calorimeter for a given weight of fuel sample. The excess of heat may amount to from 30 to 40 per cent of that from the combustion proper, and is corrected for by multiplying the heating value of the combustion as determined by a percentage factor.

Fig. 356 shows the general relation of the parts of this calorimeter, while Figs. 357 and 358 give the details of the so-called cartridge *D*. In this cartridge there is placed the prepared coal together with the proper amount of the chemical used for the generation of the necessary oxygen, the charge being inclosed gas tight. The mixing is done by shaking. After filling, the cartridge is placed in the calorimeter vessel, the covers are put on, and the fuel is lighted. Stirring is done by continuous rotations by means of the pulley *P*, Fig. 356, the water being agitated by the small blades shown at the sides of the cartridge. The casing *E* in the same figure is open top and bottom to promote water circulation. Ignition is produced in two ways, depending upon the construction of the cartridge. In Fig. 357, depressing the cap *O* against the spring *N* opens the valve *M*, and ignition may then be effected by dropping a piece of red-hot wire down the central hole. Fig. 358 shows a modification in which an electric current is used to fuse the wire *G*. Contact is made through *K* and the stem *B*. In either case the heat of ignition must be corrected for

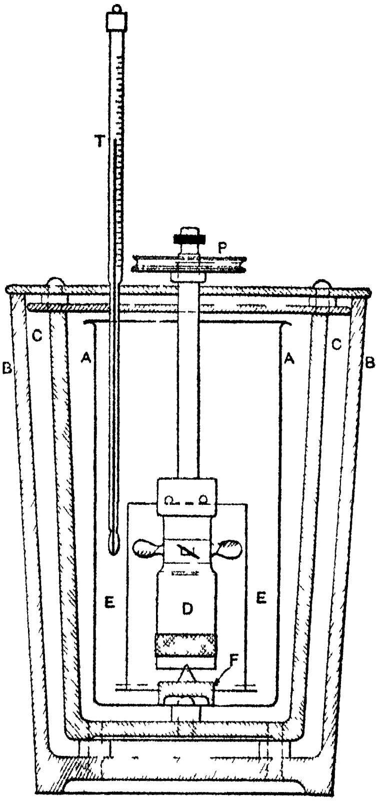


Fig 356 —Parr Fuel Calorimeter

The observations taken during a run are the same as for the ordinary bomb and the computations are made in the same way. The calorimeter allows of very quick determinations. There are, however, several things tending to impair accuracy that should be mentioned.

(*a*) The sodium peroxide combines violently with water, even in the form of moisture; it is hard material to handle, and it is likely to deteriorate in keeping.

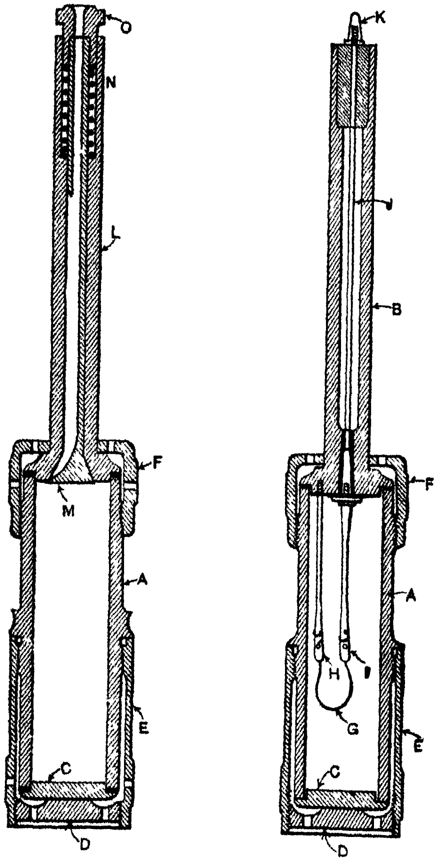


FIG. 357. FIG. 358.
DETAILS OF CARTRIDGE, PARR CALORIMETER.

(*b*) When the Na_2O_2 has aged, partially changing to NaOH, the character of the by-reactions and the heat evolution from them change.

(*c*) The excess heat from the by-reactions does not have exactly the same ratio for C, H, and S; hence a change in the composition of the fuel tested involves a change in the ratio between true heat of combustion and heat evolved in the calorimeter.

(*d*) It is hard to get complete combustion. The fused coke remaining in the " bomb " after a run frequently contains unburned material.

Despite these disadvantages, the Parr calorimeter is accurate enough for most commercial work, and when checked occasionally against a good bomb calorimeter it is quite satisfactory. This checking enables the operator to make corrections for the errors above, as they are occurring in his own work.

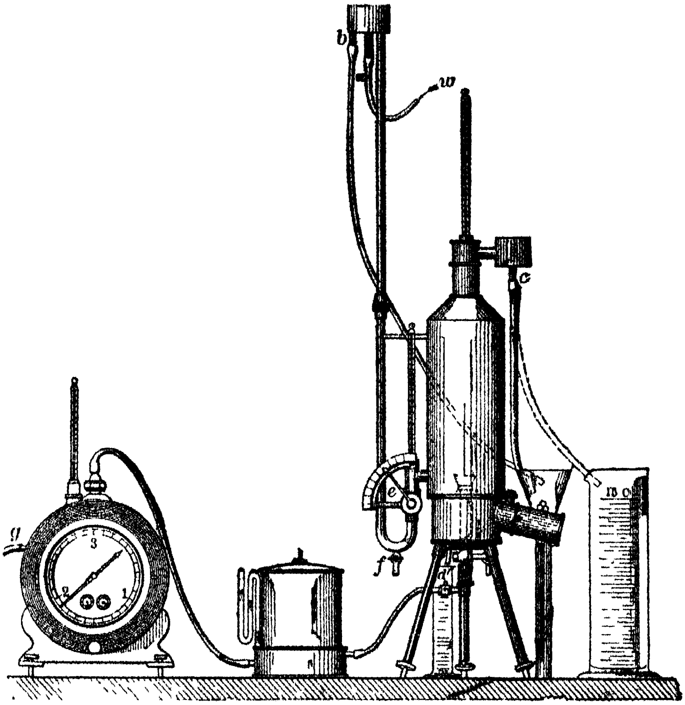


FIG. 359.—JUNKER GAS CALORIMETER.

Of the discontinuous calorimeters, the "bomb" and the Favre and Silbermann may be called " primary " instruments, being of sufficient accuracy for scientific as well as commercial work; the Carpenter, Parr, and others like them, are "secondary" instruments, good enough for most commercial work and quite accurate provided they are occasionally checked against a primary instrument.

261. Continuous Calorimeters. — *Continuous calorimeters* may be typified by the one most important in engineering, the Junker calorimeter, Fig. 359 and Figs. 360a and 360b.

Fig. 359 shows the parts belonging to a complete outfit. At the left there is a delicate wet gas meter (inlet at *g*). The gas flows through a pressure regulator and is then supplied to a burner located centrally in the calorimeter. Water is supplied at *a* (*b* being an overflow), regulated at *e*, and escapes at *c*. *f* is a drain cock. The water is measured, either in a graduated cylinder as shown, or by some other means. The water of condensation is caught and measured at *d*.

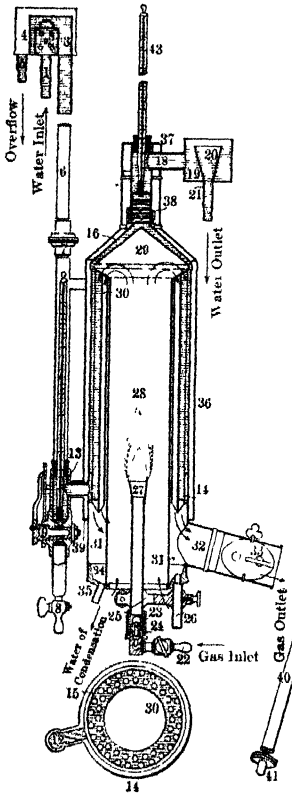


FIG. 360a.

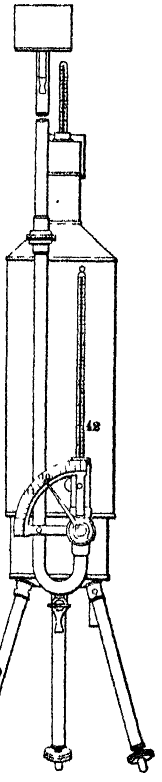


FIG. 360b.

The cross section of Fig. 360a makes the construction of the calorimeter clear. The rate of supply of water is regulated by means of the plug cock 9, and the pointer and scale 11, Fig. 360b. Temperature of the inlet water is taken at 13. The water rises in the jacket around tubes (see cross section) through which the products of combustion formed in the space 28 pass downward. The warm water passes plates 38, having staggered holes to promote thorough mixing, and flows out through 18, 20, and 21 either to the measuring apparatus or to waste. Temperature of outlet water is measured by thermometer 43. The gas from the regulator is furnished to the burner at 22. Orifice 24 can be exchanged to suit the kind of gas. Slide 23 serves to regulate the air supply. The gases of combustion formed in 28 rise to 29, pass down through 30, and escape through 31 and 32. The damper 33 serves to control the rate of escape. The pipe 32 is furnished with a thermometer to indicate the temperature of the outlet gases. The degree of cooling is thorough enough to partially condense the water vapor in the gases, the condensation flowing out through 35. The entire calorimeter body is surrounded by a polished metal jacket 36 forming an air space 39.

This instrument is adapted for either gaseous or liquid fuels, a special attachment (weighing balance) being furnished for the latter. By the device of maintaining a constant head upon an orifice the rates of flow of heat-absorbing liquid (water) and fuel (gas or liquid) are kept very closely constant. The products of combustion flow counter to the water, so that they are thoroughly cooled. By controlling the rates of flow of gas and of water as described above, the observer should be able to get simultaneously: (1) discharge gases of the same temperature as the atmosphere or the entering gases; (2) a sufficient range of temperature of the water between intake and outlet to give accuracy in the heat calculations.

The observations are:

(1) Rate of flow of water.
(2) Temperature of water entering the calorimeter.
(3) Temperature of water leaving the calorimeter.
(4) Rate of flow of fuel.
(5) Temperature of fuel.

(6) Temperature of room.
(7) Temperature of discharge gases.
(8) Amount of water of condensation.

(6) and (7) enter only into the regulation of the instrument. (1), (2), (3), (4), and (8) are used in the calorimetric calculations.

(1) is measured by observing, with a stop watch, the time required to fill a large graduate (2000 c.c., for example) with discharge water from the calorimeter. By suitable means of measuring water the operation of the calorimeter may be continued for a considerable time. By the use of an accurate scale of considerable capacity it is easy to make runs lasting several hours, and long-time trials on producer gas plants, for instance, may thus be covered by an almost continuous heating value determination of the gas. The temperature of this water is known (measurement 3), so correction can be made for its density and specific heat.

(2) and (3) are the averages of frequent readings of the water thermometers during the progress of the run.

(4) If gases are being burned, the rate of flow of gas may be measured either by the fall of a gasometer or, more commonly, by the use of a special gas meter. In either case the temperature and pressure of the gas must be taken at the point where its volume is measured. With the gas meter one has to calibrate the meter carefully — the ultimate recourse must be the gasometer. To measure the rate of flow of liquid fuels, an ingenious device is furnished with the calorimeter, consisting of a pressure tank, holding the fuel and supplying the burner, this tank being suspended from one knife-edge of a balance. The rate of flow is found by taking with a stop watch the time between throws of the balance beam when a known weight has been taken from the opposite pan of the balance between the throws.

To find the heating value of the fuel, multiply the volume of water flowing per unit of time (the time unit is arbitrary) by the density and specific heat of water at the discharge temperature, and by the temperature range of the water between intake and discharge; divide by the volume or weight of fuel flowing in the same unit time. The result is approximately the higher heating value of the fuel.

Radiation is small and is neglected. There is no water equivalent of the calorimeter to consider. If the temperature of the discharge gases differs from room temperature the results will be in error, but the amount of correction is uncertain and cannot be computed. The volume of a gaseous fuel must be corrected to whatever conditions have been determined upon as standard. In most industrial reports this standard condition must be the condition at the gas meters according to the measurements of which charges are made for gas. The results should also be given calculated to the scientific standard conditions of 32° F. and 29.92 inches of mercury pressure in order to be comparable with measurements made in other places.

The Junker calorimeter measures nearly the *higher* heating value of the fuel. It does not quite do so, because moisture escapes uncondensed as "humidity" in the exhaust gases. If the air and gas supplied to the calorimeter be both saturated with moisture, then the calorimeter will truly give the higher heating value of the fuel, for it will condense all moisture found from the combustion of hydrogen. The *lower* heating value of the fuel can be readily found under any conditions from the instrument as it is run ordinarily. The condensed moisture can be caught and measured as it drips from the instrument. Find the rate of this drip, and calculate by the methods given above, page 467, in the discussion of the difference of higher and lower heating values, the corresponding heat; subtract this from the heat measured, and the *lower* heating value of the fuel will be accurately determined. If the hydrogen content of the fuel be known, the higher heating value can be easily calculated from the lower. The true higher heating value of the fuel will be found to be considerably above the value measured by the calorimeter under the ordinary running conditions.

For ordinary gases the Junker calorimeter uses a form of Bunsen burner. Gases of low heating value, such as producer gas, can be mixed with air and burned in a simple metal tube. In burning liquid fuels, such as gasoline, kerosene, or heavier oils, a regenerative burner may be used. Air pressure forces the liquid through a coil of metal tubing which lies above the flame and is heated by it. In this coil the liquid is gasified. It then passes downward, turns upward again, and escapes through a tiny orifice in a gaseous jet, which burns much after the fashion of a Bunsen flame. In dealing with heavy oils the regenerator coil must be artificially heated before the burner can be started. Care must be used not to choke the orifice of the burner.

Commercial Fuels.

262. Composition of Fuels. — Fuels are classified as solid, liquid, and gaseous. By far the most important of all fuels is coal, the various grades of which are again classified, usually in three main classes: anthracite or hard coal, bituminous or soft coal, and lignite. Next to coal in properties come peat and wood, and vegetable and animal wastes, such as bagasse, garbage, etc. The most important liquid fuels are petroleum or petroleum products. The gaseous fuels are natural gas and various forms of artificial gases, such as illuminating gas, water gas, producer gas, blast-furnace gas, etc.

Wood and similar vegetable materials consist of cellulose, starch, water, and secondary amounts of various carbohydrates other than cellulose and starch. When decomposed by heat, out of contact with air, wood yields acetic acid, methyl alcohol, and small amounts of CH_4, H_2, CO_2, CO, C_2H_2, etc. Acetic acid and methyl alcohol are both hydroxyl derivatives of methane, CH_4. When vegetable matter is deposited in damp places, such as swamps and marshes, and slowly decomposes, it gives off CH_4, which is commonly known as "marsh gas." *Peat* is the result of decomposition of vegetable matter under water. When dried it acts as a combustible intermediate in grade between wood and coal. *Coals* are the result of long-continued slow decomposition of vegetable matter deposited in marshy places, followed by long periods of high pressure, and in some cases of quite high temperature while buried underground. Nearly all of the moisture of the peat has been driven out by pressure, and both pressure and temperature have assisted in the decomposition of the carbohydrates of the original vegetable matter. Coals consist of elementary carbon, together with hydrocarbons and carbohydrates, mostly of the methane or "paraffin" series, moisture, and such clay and sand as were deposited in the original vegetable matter, and now make the "ash" of the coal. Where the coal during its formation geologically has been highly heated, little of the hydrocarbons or carbohydrates are left, and in extreme cases the elementary carbon has been partially transformed into graphite.

Classification of coals is difficult because of the variety of the original vegetable materials, of differences in the geologic ages, and of the variations in pressures and temperatures which have played parts in determining the final result. There is hardly a definite break in the series from wood to graphite. Roughly, the parts of the series run as follows:

1. *Wood.* — Fresh vegetable material, cellulose, starch, and water; needs usually to be dried for satisfactory use.
2. *Peat.* — Vegetable matter more or less decomposed under water. Must be dried either by weathering or by pressure, or both, before use.
3. *Brown lignite.* — Carbon, complex paraffin hydrocarbons and carbohydrates, water. Loses large amounts of " hygroscopic " water by weathering (20 to 30 per cent by weight of the lignite as mined).
4. *Black lignite.* — Like the brown lignite, but with more elementary carbon and less water.
5. *Semi-bituminous coal.* — Intermediate in grade between black lignite and bituminous proper.
6. *Bituminous coal.* — Carbon, complex hydrocarbons with some carbohydrates, and no hygroscopic water.
7. *Semi-anthracite coal.* — More elementary carbon than 6, and no carbohydrates, less of hydrocarbons, and these simpler.
8. *Anthracite coal.* — Elementary carbon and simple hydrocarbons (CH_4).
9. *Graphitic anthracite.* — Very little hydrocarbons left, and some of the elementary carbon transformed to graphite by heat.
10. *Graphite.* — Final result of long-continued heat.

From peat downward through the series to graphite, moisture and ash (mineral matter) are of course present. The amounts of moisture and ash are almost entirely accidental, and have nothing to do with the above classification, which deals with the combustible portion of the fuels.

Liquid fuels, or oils, are either petroleum or petroleum derivatives. American petroleum of the eastern fields is largely a mixture of liquid hydrocarbons of the methane or paraffin series, of the general formula C_nH_{2n+2}. The first member of this series, methane, CH_4, is a gas prominent in natural gas. The heavier members of the series are liquids. By distillation these different hydrocarbons may be more or less completely separated from each other. Simultaneously with the distillation more or less chemical break down goes on, and towards the end of the distillation this chemical breakdown of the heavier hydrocarbons is the most important part of the process.

The Texas petroleum contains both paraffins and olefins, the latter hydrocarbons of the general formula C_nH_{2n}. The presence of the olefins makes the Texas petroleums less valuable as a base for the making of gasoline or kerosene, and causes the Texas oil to be used more largely in the crude state. California petroleum has an asphaltum base, and is largely used in the crude state as well as in the refined. Russian petroleum is composed mostly of olefins and therefore refines differently from the Eastern American petroleum.

Natural gas has considerable use as a fuel and illuminant in those localities favored by its occurrence. *Artificial gaseous fuels* are made by the distillation or partial combustion of coals. *Coal gas* or *illuminating gas* is a distillation product. *Water gas* combines distillation with partial combustion in the presence of steam. *Producer gas* is made by a partial combustion process in the presence of a mixture of air and steam. The characteristics of the various gases are illustrated in the following table:

Gas Variety	Volumetric Analysis						Per Standard Cubic Foot.	
	$CH_4+C_2H_4$	H_2	CO	O_2	N_2	H_2O	Weight, Pounds.	B t u. (Higher).
Natural	**93**	2	0.5	0.4	3.5	....	0.45	1100
Coal	**44**	**46**	6	0.5	1.5	1.5	0.32	730
Water	2	**45**	**45**	0.5	2.0	1.5	0.45	325
Producer	2	12	**27**	0.3	**57**	......	0.65	140

Analysis of Fuels.

The analysis of fuels is commercially necessary as a basis of determining the market value of the fuel, or its relative value for a particular case. In the following we shall consider almost entirely the analysis of coals, bringing in the oil and gas analyses incidentally in other sections of this chapter. See also Chap. XXI.

263. Coal Analysis. — There are two methods of coal analysis in common use. The *chemical* or *ultimate* analysis determines the percentages by weight of carbon, hydrogen, nitrogen, oxygen, sulphur, and ash (and generally the moisture). This kind of analysis is rather costly and is not necessary for ordinary engineering work, unless close determinations of the efficiency of the apparatus using the fuel must be made. The *engineering* or *proximate* analysis (not *approximate*, but quite as definite an analysis as the chemical) breaks the coal up into parts in the same way as it acts in the furnace, determining moisture, volatile combustible, non-volatile combustible (called fixed carbon), and ash, and also determining heating power. An incidental determination of coking power is always made. The pertinence of the proximate analysis to engineering work is obvious. It tells at a glance how the fuel will act in the furnace. If the volatile combustible percentage is high, a large combustion chamber is necessary in the furnace, with provision for oxygen supply to mix with the volatile matter and burn it. If a furnace has a relatively small combustion chamber it must, within limits, to be efficient, burn a coal with low volatile, i.e., an anthracite.

264. Sampling. — The first step toward the analysis of a coal is to obtain a truly representative sample. In sampling a carload or a pile of coal, a shovelful of coal should be taken from each of 10 or 15 places in the heap, choosing each shovelful to be representative of the material in its vicinity. In sampling the coal used in a test, of a boiler for instance, a shovelful of coal should be set aside at short intervals throughout the test from the lots of coal just before they are weighed out to be fired. In one or the other of the above ways there will be obtained a fairly representative sample of from 100 to 200 pounds' weight. This sample should be protected always from gain or loss of moisture, and from gain of foreign matter, such as dirt. This first sample is too large for laboratory use. All lumps should be broken down, until the large sample is uniformly chestnut size or smaller. Then thoroughly mix the sample and quarter it. Again crush the coal, this time to about pea size, mix thoroughly, and from this material take out the laboratory samples, *immediately sealing them air and water tight in glass jars.* Ordinary fruit jars are very convenient for this purpose. The best practice would take duplicate samples. These samples are to remain sealed air and water tight until opened in the laboratory for analysis.

In the laboratory the first act is a further grinding and mixing of the sample. A large coffee mill is a convenient machine for this. First mix the sample, then put through the mill a small part of the sample, and throw away what comes from the mill. This thoroughly cleans the mill. Grind, mix, and regrind the sample, until it is reduced to about the fineness of granulated sugar. The finer the grinding and more thorough the mixing at this stage of the process, the easier and more satisfactory will be the determinations to follow.

265. Moisture is determined by a drying process. A sample of ten or twelve grams' weight is taken, carefully weighed, then dried for an hour or so at 105° C. or 220° F. The temperature must be as high as this for thorough drying, and must not go much higher on account of the danger of driving out volatile hydrocarbons as well as moisture.* Commercially the loss of weight at the end of an hour's drying at 220° F. is called the moisture. Accurately, weights should be taken at intervals, to determine the end point of the moisture loss. The samples should be cooled in a desiccator before weighing on the chemical balance; if weighed hot, air currents in the balance chamber will vitiate the weighings. Determinations of moisture should always be made in duplicate, and should check within one-half of one per cent.

The drying cabinet may be of the simplest construction, - merely a metal box heated externally by a Bunsen burner, and provided with a thermometer by which to watch the temperature. If analyses are to be made more than occasionally, it is economical to have a more elaborate apparatus, with a thermostat to control the temperature automatically.

266. Volatile Matter is determined in a manner similar to the moisture determination, but with the temperature at red to white heat, and with careful exclusion of air in order to prevent any burning of the sample. The determination really made is (moisture + volatile), both being expelled by the process. The volatile is then found by difference, since the moisture is already known from the preceding test. A convenient method is as follows: Weigh out accurately on a chemical balance 10 to 12 grams of sample. Put it into a porcelain or fire-clay (Hessian) crucible of a capacity two or three times the volume of the sample. If the coal is not very moist, wet it slightly after putting it into the crucible. The extra moisture so introduced serves to expel air from the crucible and prevent all oxidation of the sample. Take a piece of asbestos paper somewhat larger than the top of the crucible; wet it, put the wet paper between the edge of the crucible and the lid, and weight down the lid. This gives a tight lid, which will act as a safety valve, allowing the escape of moisture or volatile from within the crucible, but preventing the ingress of air. Heat the crucible with a blast lamp capable of taking the temperature well up toward a white heat. Heat slowly for the first five minutes, while expelling the moisture; then heat to the limit of the apparatus for the balance of a half hour. Cool with the lid on. When cool, remove the lid and inspect the contents of the crucible as to coking power of the coal, noting whether the coal has coked into a single spongy mass or is still granular. Carefully remove all the residue from the crucible and weigh the residue; the loss from the original weight of sample taken is moisture + volatile matter. Run the determination in duplicate; the results should check to within one per cent. Find the volatile by difference of the result of this and the moisture determination.

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* With American anthracite and bituminous coals the temperature may go up to 300° F. without serious danger of loss of " volatile," but trouble will be met from slow oxidation of the coal being dried.

As variations on the above method the weights might be taken of crucible, crucible + sample, and crucible + residue; and special crucibles for these determinations may replace the common porcelain or fire-clay laboratory crucible. Fire-clay crucibles change their weight when first used in this test, absorbing volatile matter and decomposing it with a deposition of carbon in the crucible walls. In case the crucible is weighed, a new fire-clay crucible should therefore be saturated by a blank run before using it for an actual determination. The special crucibles which can be had for this determination have clamp lids and allow the recovery and analysis of the volatile matter and moisture driven off from the coal. This latter analysis is not of much importance commercially, save where the coal is to be used for making gas.

267. Determination of Fixed Carbon, or non-volatile combustible, is by difference. Above have been given the determination of moisture and volatile; with ash and fixed carbon added, the sum is 100 per cent. The fixed carbon is found by subtracting from 100 per cent the sum of moisture, volatile, and ash.

An explanation should be given of the relationship between the "volatile" and the "fixed carbon" found in this method of testing. The volatile represents only a portion of the hydrocarbons or carbohydrates of the coal. The heating of the coal, either in the actual furnace or in the crucible in the volatile determination, decomposes the hydrocarbons partially and the carbohydrates almost completely, leaving carbon from them to add to the elementary carbon already present in the coal, increasing the fixed carbon, and giving off as volatile simpler hydrocarbons and those in smaller quantities than the hydrocarbons contained in the coal. It is because of this decomposition of the hydrocarbons and the carbohydrates that the analysis of the volatile does not give the composition of the hydrogen-carbon-oxygen compounds in the coal.

The volatile also includes sulphur from either S or FeS_2 in the coal, CO_2 from the decomposition by heat of carbonates in the coal, and N_2O_5 from the decomposition of organic nitrates.

The smaller the quantities of hydrocarbons or carbohydrates present, the higher is the temperature required to begin their destructive distillation. In the hardest anthracites even CH_4 is decomposed and the volatile obtained is largely hydrogen, together with sulphur and oxygen and nitrogen from the decomposition of organic nitrates. Up to 20 per cent volatile in total combustible, the CH_4 probably makes up all of the hydrocarbon volatile in the coal. Beyond 20 per cent volatile in the combustible, that is, with bituminous coals and lignites, heavier hydrocarbons are present. Carbohydrates characterize the lignites. In the case of lignites and semi-bituminous coals, even heating to drive out moisture may drive out some volatile combustible.

268. The Ash Determination is best made by the quiet burning of a weighed sample in an atmosphere of oxygen at low pressure. Under high oxygen pressure the burning is too nearly explosive and ash or combustible may be blown out of the crucible. Burning in air is too slow and too liable to be incomplete for a satisfactory commercial method. In finding ash by burning in air, the sample must be heated nearly white hot in an open crucible for some hours. The burning in oxygen is complete in ten or fifteen minutes. The weights are: (1) crucible; (2) crucible + sample; (3) crucible + ash. The crucible and ash should be heated to dry them and then cooled in a desiccator before weighing. The ratio of ash weight to sample weight gives the percentage of ash. Duplicate determinations should be made and should agree to within one per cent.

Oxygen is necessary for the calorimetric determination of the heating value of coal. It is very easy to rig up a small oven or combustion chamber with an igniter, for the making of combustions in an oxygen atmosphere for the ash determination. All drafts across the crucible must be avoided. The oxygen supply must be under control so as to regulate the rapidity of combustion. Coals with high volatile must be made to burn slowly. The combustion chamber should have a window through which the burning can be watched.

The ash of a coal is not identical with the mineral matter in the coal. Coals carry sulphur and iron as FeS_2 (pyrites); sulphur as S; and frequently mineral matter as carbonates. The ash consists mostly of oxides and silicates, and is in part at least a product of the combustion of the coal. Combustion makes S into SO_2; FeS_2 into Fe_2O_3 and SO_2; carbonates into CO_2 and oxides. The CO_2 from carbonates and S from FeS_2 or S appear as volatile in the volatile test. It is therefore not scientifically correct to figure true coal by subtraction of moisture and ash from coal as received; although in general this computation is good enough as a first approximation.

269. Determination of Heating Value.—The last determination of the proximate analysis is that of the heating value. For most calorimeters, except the Parr, which requires absolutely dry fuel, it is best to use either coal as received or an air-dried sample of which the moisture loss from coal as received is known. To prepare the coal for the calorimeter, grind the coal with a mortar and pestle or equivalent device to about the fineness of flour, so that half or more is fine enough to pass a 100-mesh sieve. Do not, however, take that part of the coal which passes the sieve, rejecting that which does not. The sample so obtained would not be a fair one. Mix up well *all* of the ground material, whether it passes the sieve or not. Bottle and cork this finely ground coal immediately after it is prepared, for if left exposed in the room it will rapidly change its moisture content.

(For the operation of the calorimeter and the computation of heating power, see the preceding part of this chapter.)

Results of the proximate analysis are reported in the following form, on which the items usually observed are indicated by a star, the rest being calculated. "Coal as received" means the sample as it came to the laboratory. "Dry coal" is coal as received less the moisture, expanding the remaining items to sum up to 100 per cent. "Combustible" is coal as received less both moisture and ash; it expands the two items "volatile" and "fixed carbon" to sum up to 100 per cent.

FORM FOR REPORTING THE PROXIMATE ANALYSIS OF COAL.

	As Received, 100%.	Dry Coal, 100%.	Combustible, 100%.
Moisture	*		
Volatile matter	*		
Fixed carbon			
Ash	*		
B.t.u. per lb.	*		

Color of ash . . . . *

Coking qualities . . . . *

General remarks:

The method of calculating across from coal as received to the other columns is obviously the division of the items in the coal "as received" column by the percentages of dry coal or combustible in coal as received. It should be remembered that the calorimeters generally find the *higher* heating value of the coal.

270. **The Chemical Analysis** of a coal is one to be made only by a chemist. It determines the percentages by weight of C, H, O, N, S, and adds frequently determinations, like those of the proximate analysis, of moisture and ash.

This analysis is necessary whenever definite heat balances must be established for any test. In this connection the most important item not found by the proximate analysis is hydrogen. Happily, Prof. L. S. Marks has shown* that the hydrogen not in moisture can be found very accurately from the results of the proximate analysis. The writer has fitted equations to Professor Marks' curves, allowing the computation of the chemical analysis of a coal to be made from its proximate analysis with sufficient accuracy for most work.

In the proximate analysis the combustible is the summation of the items "volatile matter" and "fixed carbon," or it is equal to coal as received less moisture and ash. Let

V = the weight per cent of volatile matter in combustible,
H = the weight per cent of hydrogen in combustible,
C = the weight per cent of volatile carbon in combustible,
N = the weight per cent of nitrogen in combustible.

Then the following equations express the Marks' curves:

$$H = V\left(\frac{7.35}{V+10} - .013\right).$$

This gives the hydrogen not in moisture for all American coals, to an accuracy of about ± 0.2 of one per cent.

For volatile carbon (carbon occurring in the volatile matter), with an accuracy of ± 2 per cent approximately:

$C = 0.02\,V^2$ for anthracite, and $C = 0.9\,(V-10)$ for semi-anthracite,
$C = 0.9\,(V-14)$ for bituminous and semi-bituminous,
$C = 0.9\,(V-18)$ for lignites.

Sulphur in the coal directly increases the value of V; hence the calculated value of C here will be too high practically by the S content of the combustible.

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* *Power*, Vol. 29, p. 928, Dec., 1908.

For nitrogen (nitrogen comes off in the volatile matter), with an accuracy of ± 0.5 of one per cent:

$N = 0.07\ V$ for anthracite and semi-anthracite,

$N = 2.10 - 0.012\ V$ for bituminous and lignite.

Oxygen and sulphur are too widely variant to allow of any calculation; their amounts are more or less accidental.

A chemical analysis sufficient for the purposes of the engineer can be obtained from the proximate analysis by the use of the above equations, as the following example will show:

Example. — The following are the proximate and chemical analyses given by the U. S. Geological Survey for a certain sample of Illinois coal:

	H_2O	Vol. Matter	Fixed C	Ash
Proximate Analysis, per cent,	12.91	31.9	43.55	11.64

	H	Total C	N	O	S	Ash
Chemical Analysis, per cent,	5.43	60 74	1.15	19.72	1.32	11.64

Evidently since the ash content of these two analyses is the same, the percentages of H and O given in the chemical analysis must include the percentages of these gases in the 12.91 per cent of H_2O given in the proximate analysis. Restating this analysis so as to have the water appear, we will have:

	H	Total C	N	O	S	Ash	H_2O
Chemical Analysis, per cent,	4.00	60.74	1.15	8.24	1 32	11.64	12.91

The problem is to see how closely this chemical analysis can be checked by starting with the proximate analysis above.

Computing the proximate analysis to the basis of combustible by dividing by $(1.0 - .1291 - .1164) = .7545$, we will have:

Vol. Matter	Fixed C
42.3	57.7

Substituting in the equations given above ($V = 42.3$) we then obtain

$$H = 42.3\left(\frac{7.35}{42.3 + 10} - .013\right) = 5.39 \text{ per cent,}$$

$$\text{Vol. } C = .9\ (42.3 - 14) = 25.47 \text{ per cent,}$$

and

$$N = 2.10 - .012 \times 42.3 = 1.59 \text{ per cent.}$$

The analysis of the combustible will now read as follows:

H	Vol. C + Fixed C = Total C	N	Rest		Total
5.39	83.17	1 59	9.85	=	100

Recomputed to the basis of coal as received by multiplying by .7545 we will have:

H	Total C	N	Rest (O)	Ash	Water
4.07	62.75	1.20	7.43	11.64	12.91

Comparing these figures with those given by the chemical analysis above it will appear that the agreement is fairly close with the exception of that for total carbon. As pointed out above, this is largely due to the fact that in this method of computation the sulphur content is practically all added to the total carbon, and if the sulphur content of the coal be known, as it is in this case (1.32 per cent), correction can be made so that the computed chemical analysis will finally show.

H	Total C	N	O	S	Ash	H_2O
4.06	61.44	1.22	7.22	1.32	11.64	12.91

271. Sulphur Determination. — A determination of sulphur is occasionally of interest, and in some cases necessary, as, for example, in coals used to make illuminating gas and for the fuel for gas producers, cupolas, or blast furnaces. High sulphur will make its presence known without special analysis. In ordinary boiler work sulphur is fairly harmless. In economizer installations, where the flue gases are cooled considerably, the SO_2 and SO_3 formed from combustion of S combine with moisture to form H_2SO_3 or H_2SO_4 (sulphurous or sulphuric acid), which corrode the joints of the economizer. Similar trouble may occur in ordinary boiler work, but there the temperature of the flue gases is usually too high to allow the formation of much of the acids.

The American Chemical Society directions for sulphur analysis are as follows:

Mix thoroughly one gram of finely powdered coal* with one gram of magnesium oxide and one-half gram of dry sodium carbonate in a thin platinum dish having a capacity of 75 to 100 c.c. A crucible may be used, but a dish is preferred. The magnesium oxide should be light and porous, not a compact, heavy variety.

The dish is heated in a triangle over an alcohol lamp, held in the hand at first. *Gas must not be used*, because of the sulphur that it contains. The mixture is frequently stirred with a platinum wire and the heat raised very slowly, especially with soft coals. The flame is kept in motion and barely touching the dish, at first, till strong glowing has ceased, and is then increased gradually till, in fifteen minutes, the bottom of the dish is at a low, red heat. When the carbon is burned, transfer the mass to a beaker and rinse the dish, using about 50 c.c. water. Add 15 c.c. of saturated bromine water and boil for five minutes. Allow to settle, decant through a filter, boil a second and third time with 30 c.c. of water, and wash till the filtrate gives only a slight opalescence with silver nitrate and nitric acid. The volume of the filtrate should be about 200 c.c. Add one and a half cubic centimeters of concentrated hydrochloric acid, or a corresponding amount of dilute acid (8 c.c. of an acid of 8 per cent). Boil till the bromine is expelled, and add to the hot solution, drop by drop, especially at first, and with constant stirring, 10 c.c. of a 10 per cent solution of barium chloride. Digest on the water-bath, or over a low flame, with occasional stirring, till the precipitate settles clear quickly. Filter and wash, using either a Gooch crucible or a paper filter. The latter may be ignited moist in a platinum crucible, using a low flame, till the carbon is burned. Weigh as barium sulphate and calculate sulphur.

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* With coals high in moisture a correction may be necessary on account of the loss of water in powdering the coal. (See above, under Moisture.)

In the case of coals containing much pyrites or calcium sulphate, the residue of magnesium oxide should be dissolved in hydrochloric acid and the solution tested for sulphuric acid.

272. Methods of Computing the Heating Value of Fuels. — The chemical or ultimate analysis of coal has been much used for a computation of the heating value of the coal, after the Dulong formula. If C represents the weight percentage of carbon in the coal, H that of hydrogen, O that of oxygen, S that of sulphur, the Dulong formula gives the following heating power:

$$\text{B.t.u. per pound} = 14{,}540\,C + \begin{Bmatrix} 52{,}500 \\ \text{or} \\ 61{,}950 \end{Bmatrix} \left(H - \frac{O}{8}\right) + 4020\,S.$$

The coefficients are the heating powers of the separate chemical elements, with that of hydrogen giving either the lower or the higher heating value according to the coefficient used. The term $\left(H - \frac{O}{8}\right)$ is supposed to contain a correction for hydrogen already combined with oxygen in the coal as moisture. The formula is in a sense rational, but the following points should be noted:

The carbon and sulphur are the only elements present in the coal in a free state, and only a portion of them is elementary. Part of the elementary carbon may be, and in some anthracites certainly is, graphite. Graphite has a decidedly lower heating value than amorphous carbon (about 14,300 instead of 14,540). Part of the sulphur is present as FeS_2 (pyrites). The sulphur may burn to either SO_2 or SO_3; and depending on its initial condition and the compound formed in burning, the heating power varies widely. Combined carbon and hydrogen in hydrocarbons have not the same heating power as if they existed separately side by side; the heat of formation or dissociation of the hydrocarbon must be considered. This makes the heating powers for part of the carbon and all of the hydrogen wrong. The amount of error depends on the particular set of hydrocarbons present in the coal; in some cases it may amount to 20 per cent of the Dulong formula value for the heating value of a hydrocarbon. The term $\left(H - \frac{O}{8}\right)$ does not contain a proper correction for the hydrogen contained in moisture, for not all of the oxygen in the coal is combined with hydrogen. Part of the oxygen is probably combined with nitrogen in organic nitrates and part may be present in carbonates in mineral matter caught in the coal.

Taking these facts into consideration, it is not surprising to find the difference between the Dulong formula values of heating power and the values determined experimentally in a calorimeter to be frequently as high as 10 per cent. Usually the formula comes within 5 per cent in the cases of anthracite coals. The error of the formula, with the coefficients given above, is usually an error of excess.

Empirical formulas, of the same type as the Dulong, but with coefficients worked out by comparison of chemical analyses with experimentally determined heating powers, give better results, but are still too widely and irregularly in error to be allowable in modern engineering work. One such formula, of fairly wide use, is that of the Verein Deutscher Ingenieure. It gives the *lower* heating value of a coal, with W = per cent of moisture, and other symbols as above.

$$\text{Calories per kilogram} = 8000\,C + 29{,}000\left(H - \frac{O}{8}\right) + 2500\,S - 600\,W.$$

$$\text{B.t.u. per pound} = 14{,}400\,C + 52{,}000\left(H - \frac{O}{8}\right) + 4500\,S - 1100\,W.$$

The only way to get the heating value of a fuel reliably and accurately is to determine it experimentally with a calorimeter. Such a calorimeter determination forms a part of the proximate analysis of a coal.

Similar empirical formulas have also been established for computing the heating value of hydrocarbon combinations, either solid, liquid, or gaseous. It will not do simply to proceed on the Dulong principle and to multiply the percentages of C and H

contained by the heating value of these elements, for reasons pointed out above. For gaseous hydrocarbons like CH_4, C_2H_2, C_2H_4, etc., the following empirical formula, due to Slaby, gives satisfactory results:

Lower heating value = $(112 + 18{,}880\,y)$ B.t.u. per standard cubic foot, in which y = the weight of a standard cubic foot of the gas in pounds. Thus for CH_4, $y = .04464$, so that the heating value is

$$(112 + 18{,}880 \times .04464) = 952 \text{ B.t u.}$$

The heating value of commercial gases, like producer gas or illuminating gas, can of course be directly determined from its percentage composition as soon as the heating value of the individual combustible gases it contains is known.

273. Buying Coal by Analysis. — The increasing use of coal analyses as a basis for determining coal values, and the growing practice of buying by analysis, make it desirable to indicate how business firms not fully equipped for the making of analyses may keep check on the quality of the coal they are buying or selling. As a general thing, coals coming from the same mine at different times, or even from different points of the same coal field, do not vary greatly in the composition or value of their combustible portions. The variations are in the ash and moisture content. By sending samples at intervals to some laboratory, complete analyses of the coal may be obtained, which will give the nature and value of the combustible part of the coal. Then the firm may make frequent checks, upon every large delivery, say, of ash and moisture. These latter determinations can be made by working with fairly large weights of samples, with no more elaborate apparatus than a fairly accurate bench scale, some tin pans, a drying oven, a thermometer, two or three Bunsen burners, and a few Hessian or porcelain crucibles, which equipment should not cost more than $20.00.

274. Flue Gas Analysis and Combustion Calculations. — The gases analyzed in engineering work are those resulting from combustion, commonly called "flue gases."* They consist of varying mixtures of nitrogen, oxygen, and carbon dioxide, with lesser amounts of carbon monoxide, hydrogen, water vapor, and sulphur dioxide. The analysis is useful as a basis from which to judge of the efficiency of combustion.

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* For the analysis of producer gas, see the chapter on the testing of gas engines and producers.

The methods to be employed must be such as any engineer can fully comprehend, and the apparatus portable and convenient. The degree of accuracy sought need not be such as would be required in a chemical laboratory where every convenience for accurate work is to be found. Indeed, considering the approximations to be made in its application, it is very doubtful if determinations nearer than one per cent in volume are required, or even of any value. Such determinations are obtained readily with simple instruments, and serve to show the approximate condition of the gaseous products of combustion. The student is referred to "Handbook of Technical Gas Analysis," by Clemens Winkler (London, John Van Voorst), and to "Methods of Gas Analysis," by Dr. W. Hempel, translated by L. M. Dennis (Macmillan Company); also to a paper on tests of a hot-blast apparatus by J. C. Hoadley, Vol. VI, "Transactions of the American Society of Mechanical Engineers."

275. Sampling. The first step in the analysis is the obtaining of a representative sample. In the flues the mixture of the gases may be far from homogeneous. Hence the sampling apparatus must either provide for mixing the gases before taking the sample at one point, or must take the sample at so many points, and points so distributed, as to get a fair average. Allowance should be made in the latter case for variation in the velocity of flow of gases in different parts of the flue. It is easy to provide baffles to cause eddy currents in the flue, and so bring about a good mixing of the gases. Then a single sampling point near the center of the flue is sufficient.

The simplest collecting arrangement is a $\frac{1}{4}$-inch pipe of such length as to reach nearly across the flue. This pipe is perforated with a number of small holes spaced along its length, the end of the pipe being plugged up. This scheme is very often used in practice, and where the mixture of the gases may be assumed to be fairly uniform it is probably satisfactory. It is open to the objection that probably a greater proportion of gas is drawn into the tube through the holes near the aspirator than through the openings farther away.

A more elaborate scheme (see "Trans. Am. Soc. Mech. Eng.," Vol. VI) consists in running a number of collecting tubes, open simply at the end, to various parts of the flue and connecting all the outside ends of these tubes to a common mixing box into which an aspirator then draws the gas and from which the sample is taken. Even this scheme may not give a uniform mixture, because with the same suction on all the tubes, the shorter tubes will certainly supply more gas to the mixing box than the longer ones.

The material for the collecting tube or tubes is preferably porcelain or glass, but iron has no specially selective or absorptive action upon any of the gases, and iron pipe is therefore often employed. Use as little rubber tubing as possible.

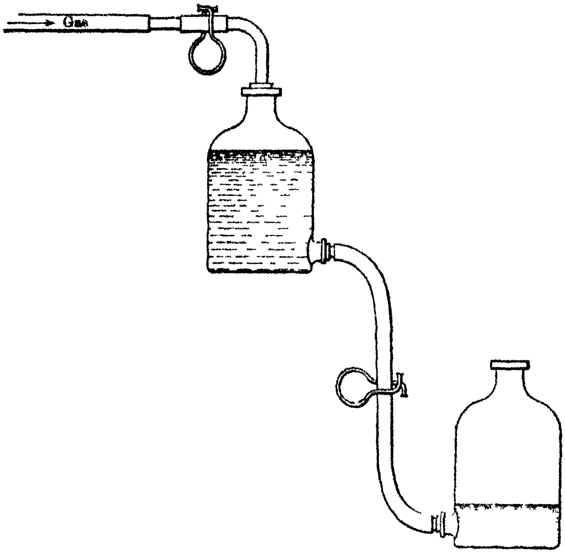

FIG. 361.—ASPIRATOR.

The gas should always be collected as closely as possible to the furnace, after combustion is sure to be as complete as it will become in the apparatus under test. At any distance away in the flue there is apt to be dilution due to inleakage of air. If the sample must be taken some distance away, the flue must be examined for leakage and the leaks stopped.

As the pressure in the flue is generally below atmospheric, it is necessary to have some form of *aspirator* to draw out the sample. The aspirator should either work continuously or should be worked for such a time before taking a sample as to secure a thorough flushing out of the system of piping used in the sampling device.

The aspirating apparatus may take several different forms. The most common type consists of two bottles with double outlets and connected as shown in Fig. 361.

The operation of this apparatus is of course intermittent, and since the bottles are usually not very large, the time during which any given sample is taken is comparatively short, which means that many of them are necessary thoroughly to cover a test.

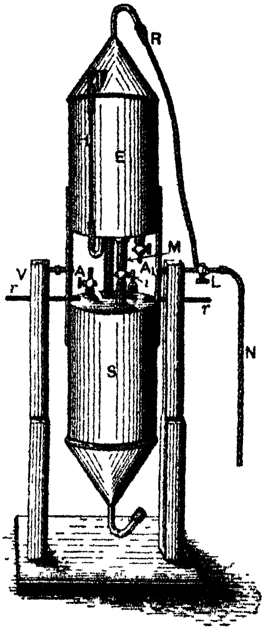

FIG. 362.

A modification of the simple bottle aspirator by means of which it is possible to take a time sample is illustrated in Fig. 362. Two cans, E and S, are mounted as shown about the shaft V. In the position shown, E is connected with the sampling tube through the connections R, L, N. The water, with which E was originally filled to the top, is allowed to drain out of E into S through the connecting pipe M, the rate of transfer being controlled by the setting of the cock, i. As the water recedes, E fills with gas. The water level is continuously indicated on the glass H. In the meantime the gas, which it is assumed was collected in S when this was in the top position, is displaced out of S through connection A and is either taken to the analyzing apparatus or wasted. When all the water is transferred, the position of the vessels is reversed and the apparatus is ready for a new sample. The cut is defective in that no stopcocks are indicated in the outlets near the conical heads of the vessels.

The transfer liquid used in collecting apparatus of the type described is usually water. It will be pointed out below that water has a certain influence upon the composition of flue gases with which it comes in contact, and it is consequently better to collect without contact with water if possible. One method of doing this is indicated in Fig. 363. This apparatus consists of a glass sampling tube furnished with glass stopcocks. These tubes can be bought from any chemical dealer. At one end the tube is connected to the flue, at the other to a rubber aspirator bulb. By alternately squeezing and releasing this bulb, gas will be drawn into the tube. Enough must be drawn through to replace the air or old gas in the tube The proceeding is therefore rather laborious, but the sample obtained will not be disturbed by contact with water. These tubes are also the best means of collecting and storing gas, if the analysis cannot be performed on the spot. The stopcocks should be greased with vaseline from time to time, both to prevent their sticking and to make them gas-tight.

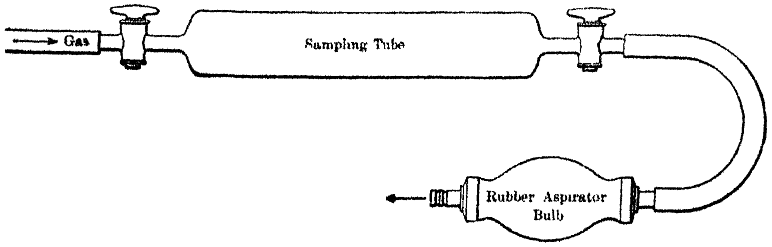

FIG. 363. —ASPIRATING AND COLLECTING APPARATUS FOR FLUE GAS.

To cut down the labor of filling the tube with fresh gas, an ejector is a handy instrument, where water or steam of sufficient pressure to operate one is available. Fig. 364 shows the construction and operation of one type.

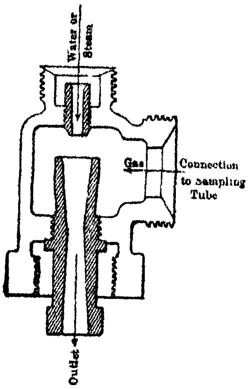

FIG. 364. — EJECTOR FOR ASPIRATION OF GAS SAMPLE.

Water is a solvent for all of the flue gases. For oxygen and nitrogen, tap water is already saturated; but it will absorb considerable quantities of CO_2, CO, H_2, and SO_2. Hence it is necessary to draw a gas sample into the sampling bottle, shake liquid and gas sample together, reject that sample, draw another, and repeat, until the water in the sampling bottle is *saturated with the particular gas mixture which is being sampled.* It is desirable not to use new water each time in the sampling bottles, but to keep and use the same liquid over and over again. This reduces the possible error from failure to saturate the water with the gases. H_2O vapor in the flue gas of course disappears partly at this stage of the sampling process. SO_2 probably does the same, and it is unlikely that SO_2 would be found in the gas in the sampling bottle. If it were it would be shown in the subsequent analysis as CO_2.

The solubility of the gases in the collecting liquid may be limited. The scientific method is the collection of the gas sample over mercury instead of water. The addition of glycerine to the water gives a liquid in which the solubility of the gases is less than with water alone. The addition of a slight amount of some strong acid, as H_2SO_4, will keep CO_2 from going into solution. The collection of samples is made more rapid, and analysis more accurate, by the addition of glycerine and H_2SO_4 to the sampling water.

276. Analysis of Flue Gas. — The analysis ordinarily employed is volumetric. A definite portion of the sample, usually 100 c.c., is taken and submitted in turn to reagents capable of absorbing the various component gases. After each absorption the volume of the remaining gas is measured. The decrease of volume from the preceding measurement shows the volume percentage of the gas taken up by the last absorbent.

The absorbents used, their order of use, and method of action, are as follows:

1. KOH (potassium hydroxide, caustic potash) in solution in water is used to absorb CO_2. It will not absorb N_2, O_2, CO, or H_2, but it will absorb SO_2 if any be present in the sample. As large a surface of the KOH solution as possible should come in contact with the gases. Hence the gases should either be made to bubble through the solution or the absorption bulb should be filled with glass beads or iron wire gauze, which will expose to the gases a large surface wet with the KOH solution. The strength of the solution is ordinarily one part by weight of KOH to two parts of water.

2. Oxygen may be absorbed by one or the other of two reagents. Neither of these reagents absorbs CO, N_2, or H_2. They are: (*a*) an alkaline solution of pyrogallic acid; (*b*) phosphorus. (*a*) To make the pyrogallic acid solution, dissolve 5 grams of pyrogallic acid in 15 c.c. of water, and 120 grams of KOH (stick form) in 80 c.c. of water, and mix the two solutions. The alkaline solution so made will absorb CO_2 or SO_2; hence these must have been removed by the KOH solution before the absorption and determination of O_2 is attempted. The use of the pyrogallic acid solution is similar to that of the KOH solution, save that it is advisable to protect the pyrogallic solution from light. (*b*) Phosphorus for oxygen absorption should be in stick form. It is kept under water. When the gas sample to be analyzed is taken into the phosphorus reagent bulb, it displaces this water; the O_2 reacts with the P, forming white clouds of fumes of P_2O_5. These clouds soon settle and dissolve in the water. The phosphorus bulb should be protected against light. To form the phosphorus sticks, melt the phosphorus under water in a beaker set in a water-bath. Heat very gently. After melting, take a glass tube having an internal diameter of the size of the sticks desired. Push this into the melted phosphorus. Place the finger over the upper end of the tube and quickly transfer the tube with the phosphorus it contains to a beaker containing cold water. If the tube has been cut off square at the lower end and is not of too large diameter, this operation can be carried out without any danger whatever. After cooling, push the stick out of the tube into cold water by means of a glass rod. The strict rule to be observed is that phosphorus should never be handled except under water or oil. The reagent tubes for the absorption of oxygen must be so constructed that they remain air-tight.

3. Carbon monoxide is absorbed in an ammoniacal copper chloride solution. The solution does not absorb N_2 or H_2, but does absorb CO_2, SO_2 or O_2. Hence it must follow the KOH and P (or pyrogallic acid) absorptions. This copper chloride solution is hard to make, does not keep well, and has a comparatively limited absorbing power. Hence the determination of CO is often omitted in ordinary flue gas analysis, although this is never advisable.

To make the solution, dissolve 10 grams of copper oxide (CuO) in 100 to 200 c.c. of concentrated hydrochloric acid (HCl), and allow the solution to stand in a stoppered flask filled with copper wire until it becomes colorless, that is, has become cuprous chloride. This clear solution is poured into a beaker containing 1½ to 2 liters of water. The cuprous chloride will be precipitated. Pour off the liquid above the precipitate, add 100 to 150 c.c. of distilled water, and add ammonia until the liquid takes a pale blue color. The solution should be used fresh. It is preserved, and its capacity for CO increased, by immersing in it spirals of fine clean copper wire.

4. Hydrogen may be absorbed and measured by (*a*) combustion; (*b*) by taking it up in palladium. (*a*) After removal of CO from the sample, H_2 is the only combustible left. Hence an amount of air, about equal to the remaining gas sample, may be taken into the measuring tube, measured, and this new gas mixture may then be passed into a combustion tube where a succession of electrical sparks, or an electrically heated wire, will cause the burning of the hydrogen, with the oxygen of the air just added. The water formed has a relatively negligible volume; therefore remeasurement gives the volume of the hydrogen and oxygen consumed. As the volumetric reaction is $2\,H_2 + O_2 = 2\,H_2O$, two-thirds of the volume diminution by the combustion is the volume of the hydrogen. (*b*) Palladium foil has the peculiar property of absorbing hydrogen gas in large amounts at ordinary temperatures, although it does not absorb other gases. The hydrogen may be driven out of the palladium by heat, and the palladium is then ready to use again. This method of absorbing hydrogen is more suited to commercial use than the combustion method. It requires, besides the absorbing tube containing the palladium foil, a small alcohol lamp or other device for heating the palladium.

5. Moisture (H_2O vapor) in flue gases can be found by drawing the gases through a calcium chloride tube, obtaining the increase of weight of the tube, and comparing this weight of H_2O with the weight of the gas drawn through, figured from volume and density. This measurement of H_2O is rarely made, as it is possible to calculate the H_2O from the analyses of the coal and of the flue

gases. If it is made it must of course be done before the gas ha come in contact with water.

277. Absorbing Power of the Reagents. — This is stated in cubi centimeters of the particular gas which one cubic centimeter of th reagent concerned will absorb. The following table, however, give the cubic centimeter of gas that may be absorbed before the reagen ceases to act with a fair degree of rapidity.

Gas to be Absorbed	Reagent.	Absorbing Power.
CO_2	KOH solution	40
O_2	Potassium pyrogallate	2 25
	Phosphorus	Practically unlimited if protected properly.
CO	Ammoniacal copper chloride solution	4 0

The full absorbing power can be obtained approximately by mult plying the factors given by 4.

278. General Forms of Flue Gas Analysis Apparatus and Metho of Operation. — The apparatus employed for volumetric gas analysi consists of a measuring tube, in which the volume of gas can b drawn and accurately measured at a given pressure, and a treatin tube into which the gases are introduced and then brought in con tact with the various reagents already described. The apparatu employed may be divided into two classes: (1) those in which ther is but one treating tube, the different reagents being successivel introduced into the same tube; (2) those in which there are as man treating tubes as there are reagents to be employed, the reagent being used in a concentrated form, and the gases brought into con tact with the required reagent by passing them into the specia treating tube.

In either case the steps are as explained in Article 276. (*a*) Ob tain 100 c.c. in the measuring tube; (*b*) transfer to CO_2 treatin tube and absorb CO_2. Transfer back to measuring tube and not the reduction in volume. Repeat the operation until there is n further reduction. If the reagent is in good shape, the absorptio should be complete after two transfers. If the reduction is measure in cubic centimeters, the reading is per cent CO_2 direct. (*c*) Trans fer remainder of gas to treating tube for O_2 and repeat the operations under (*b*). The total reduction in volume will now be the sum of $CO_2 + O_2$. (*d*) Transfer to all the other reagent tubes in their proper succession in the same manner.

The following form is convenient for the recording of data, and also shows the principal items of computation based upon the analysis.

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

FLUE GAS ANALYSIS.

Location of Plant.......................... *Date*...... ....19....
Owners.......... .... *By*................ ......
Type of Boilers ....
Number of Boilers...... . .. .
Character of Draft.... .. ..

	Determination Number.	1	2	3	4	5	6
	Time of sample						
	Draft gauge, inches water						
	Temperature of flue, deg. F.						
	Temperature of boiler room, deg. F.						
Per cent by Volumes	CO_2						
	$O_2 + CO_2$						
	Free oxygen						
	$O_2 + CO_2 + CO$						
	CO						
	Nitrogen						

Based on Analysis of Flue Gas Sample No.	Per cent by weight, CO_2		
	" " " O_2.		
	" " " CO		
	" " " N_2		
	Weight flue gas per 100 cu ft		
	Weight free oxygen in 100 cu. ft. flue gas		
	Weight air per 100 cu. ft. flue gas		
	Weight carbon per 100 cu. ft. flue gas		
	Weight air per pound carbon		
	Weight air per pound coal		
Ultimate analysis of coal. Per cent by weight.	Carbon		
	Hydrogen		
	Oxygen		
	Sulphur		
Theoretical air, pounds per pound coal			
Ratio, actual to theoretical air supply			
Heat units lost per pound coal			
Heating value of coal			
Per cent of heat lost in flue			

In performing these various operations it is essential that the tubes be kept clean and that the reagents be kept entirely separate from one another. This is accomplished by washing or causing some water to pass up and down the tubes or pipettes several times after each operation.

Flue gas analysis apparatus containing only one treating tube is not very often used at present. To this class belongs the Elliot apparatus described below. The other class, that having separate treating tubes for each gas to be absorbed, is exemplified by two main types, the Orsat and the Hempel. There are a number of modifications of the original Orsat apparatus, but the changes are only in details. The Hempel apparatus can be used for the analysis of any gas, while the Orsat is so arranged that the maximum reduction of volume that can be accurately read is about 21 or 22 per cent, this being in any flue gas the maximum of the sum of $CO_2 + O_2 + CO$.

Of all the apparatus mentioned, the Hempel is probably the most reliable and is the one generally employed by chemists in the laboratory. Unfortunately it is bulky and cannot be easily arranged as a traveling set.* For this reason engineers generally prefer the Orsat, which when arranged in a traveling case does not need to occupy more space than 6 inches by 15 inches by 24 inches, the latter dimension being the height. If carefully handled, this apparatus will give results which are satisfactory in most engineering work. As between the Hempel and the Orsat, the claim is made that certain of the gases, particularly CO, will not absorb completely without active shaking together of reagent and gas. This procedure is of course not possible in the Orsat as commonly constructed. Objection is also made to the volume of gas contained in the long capillary tubes usually used, which volume may be a considerable percentage of that really measured.

279. Elliot's Apparatus. — This is one of the most simple outfits for gas analysis, and consists of a treating tube *AB* and a measuring tube *A'B'*, Fig. 365, connected by a capillary tube *E* at the top, in which is a stopcock *G*. The tubes shown in Fig. 365 are set in a framework having an upper and a lower shelf, on which the bottles *L* and *K* can be placed. In using the apparatus, it is first washed, which is done by filling the bottles with water, opening the stopcocks *F* and *G*, and alternately raising and lowering the bottles *K* and *L*. The bottles and tubes are then filled with clean distilled water, raised to the positions shown, and the stopcocks *G* and *F* are closed. The gas is then introduced by connecting the discharge from the aspirator to the stem of the three-way cock *F*, turning it so that its hollow stem is in connection with the interior of the tube *AB*; lowering the bottle *L*, the water will flow out from the tube *AB* and the gas will flow in. When the tube *AB* is full of gas the cock *F* is closed, the aspirator is disconnected, and the gas is measured in *A′B′*. The gas must be measured at atmospheric pressure. That may be done by holding the bottle *K* in such a position that the surface of the water in the bottle shall be at the same height as that in the tube. A distinct meniscus will be formed by the surface of the water in the tube; the reading must in each case be made to the bottom of the meniscus. To measure the gas, which will be considerably in excess of that needed, the cock *G* is opened, the bottle *K* depressed, the bottle *L* elevated; the gas will then pass over into the measuring tube *A′B′*; the bottle *K* is then held so that the surface of the water will be at the same level as in the measuring tube, and the bottle *L*

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* Owing to the efforts of Prof. L. M. Dennis of the Dept. of Chemistry, Cornell University, the firm of Greiner und Friedrichs, Sturtzerbach i/Th. is now engaged in designing a traveling "Hempel," making smaller burettes and pipettes without serious loss of accuracy.

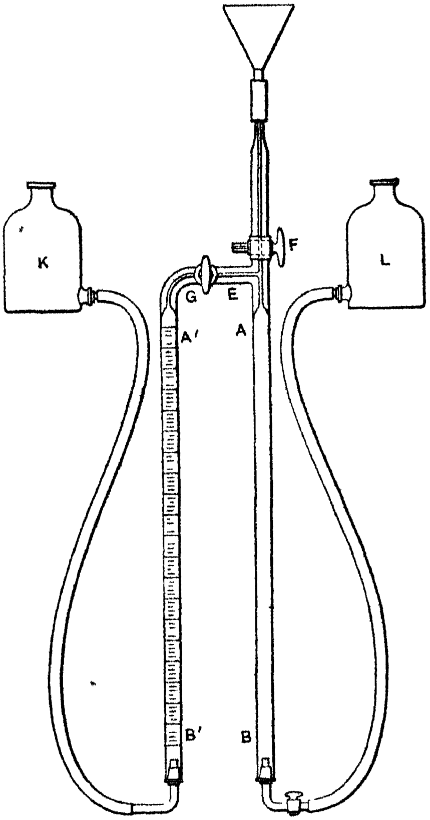

Fig 365.—Elliot Apparatus for Flue Gas Analysis.

manipulated until exactly 100 c.c. are in the measuring tube; then the cock *G* is closed, the cock *F* opened, the bottle *L* raised, and the remaining gas wasted, causing a little water to flow out each time to clean the connecting tubes. The measuring tube *A'B'* is surrounded with a jacket of water to maintain the gas at the uniform temperature of the room. After measuring the sample it is then run over into the treating tube *AB*, and the reagent introduced through the funnel above *F* by letting it drip very slowly into the tube *AB*. After there is no further absorption in the tube *AB*, the cock *F* is closed and the gas again passed over to the measuring tube *A'B'* and its loss of volume measured. This operation is repeated until all the reagents have been used; in each case when the gas is run back from the measuring tube, pass over a little water to wash out the connections; exercise great care that in manipulating the cocks *F* or *G* no gas be allowed to escape or air to enter.

280. The Orsat Apparatus. — The arrangement of a complete Orsat is shown in Fig. 366.* The gas is drawn through *k* into the measuring tube *a* by manipulating the bottle, after the old gas has been driven out through *k* by raising the bottle. The gas is brought to atmospheric pressure first by drawing in a little gas in excess and closing the stopcock *k*, then by raising the bottle the gas is slightly compressed and the excess may be driven out by a quick turn of *k*. Repeat this until, with *k* closed, the water in the measuring tube will stand at zero when the levels in the tube *a* and in the displacement bottle are at the same height. The apparatus should then contain just 100 c.c. at atmospheric pressure, but if this volume includes everything up to the cock *k*, it will be seen that not all of the gas can at any time be transferred into the reagent tube *b*, for instance. This points out the nature of the error previously mentioned in connection with the capillary tube. The error is minimized as far as possible by having the liquid in the reagent tubes stand above the stopcocks up to marks put on the glass near the juncture with the horizontal capillary when the measurement of gas is made, either initially or at any time during the analysis. But this requires extremely careful handling, otherwise some of the reagent from one tube is sure to be run into the next or into the measuring tube *a*.

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* Reproduced from Hempel's "Gas Analysis," L. M. Dennis.

The method of operation has otherwise been explained, see Articles 276 and 278. *b* is the reagent tube for CO_2, *c* that for oxygen, *d* that for CO, and *f* the palladium asbestos tube for hydrogen; *g* being an alcohol lamp for driving the hydrogen out of the palladium after the analysis. The tube *f* is rarely used in engineering analyses.

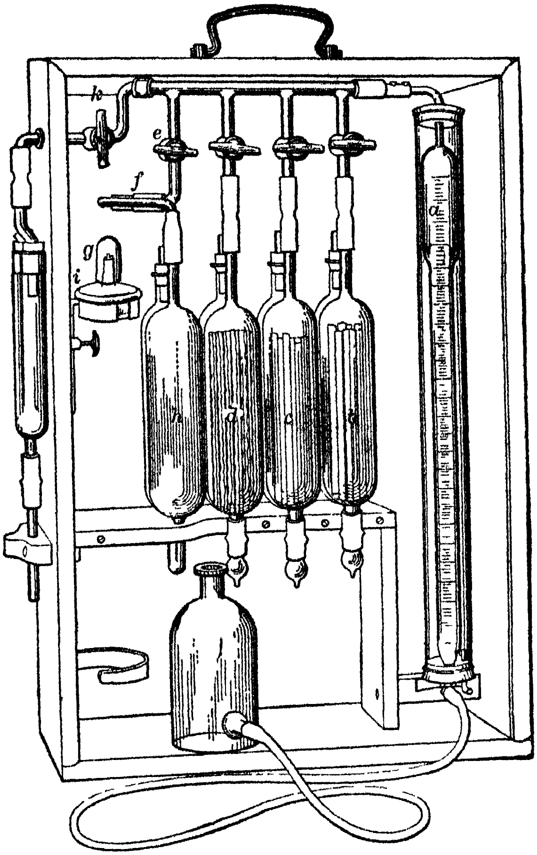

FIG. 366.—ORSAT APPARATUS FOR FLUE GAS ANALYSIS.

281. The Hempel Apparatus for Gas Analysis. — This consists of (*a*) the measuring burette and (*b*) absorption pipettes of different forms fitted to the various reagents used.

(*a*) The measuring burette is shown in Fig. 367. It consists of a leveling tube *a* and a measuring tube *b*. Both are firmly fixed in cast-iron supports, as shown, and are connected across at the bottom by rubber tubing. The measuring tube *b* is furnished with stopcocks *d* and *c*, and the volume between them is made exactly 100 c.c. The tube has a scale *E*, the smallest division of which is $\frac{1}{5}$ c.c. The transfer liquid used is either mercury or water, usually the former. To fill the burette, open *d* and *c* and raise *a* until the liquid appears at *c*, then close *d*. Now connect to the source of gas supply, lower *a* and open *d*, when the gas will be drawn in. To measure the gas, either initially or at any time during an analysis after it is transferred to the burette, close *d*, and, bringing *a* and *b* together, raise or lower *a* until the two levels are at the same heights. This puts the gas in the tube under atmospheric pressure and determines the volume at that pressure.

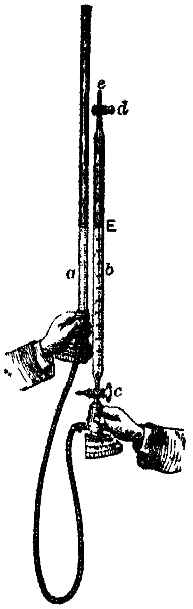

FIG 367.

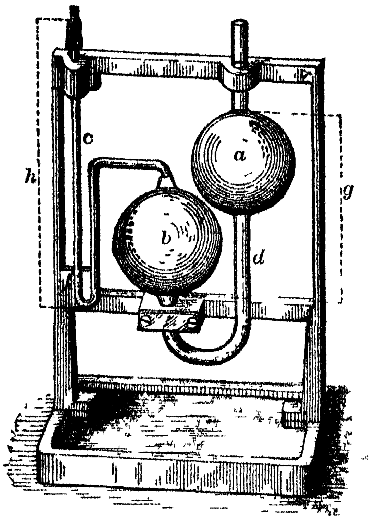

FIG 368 —HEMPEL ABSORPTION PIPETTE

(*b*) These are of various forms, depending upon the nature of the reagent employed. Fig. 368 shows a form of simple absorption pipette which may be used for liquid reagents which do not deteriorate on contact with air. A modified form is also made for use with solid reagents like phosphorus. Connection to the measuring burette for the transfer of gas is made to the tube *c*.

The second form is a double pipette and is used for reagents like alkaline pyrogallol, etc., which soon spoil when kept in contact with air. Fig 369 shows a form which can be used with both liquids and solids. Connection to the measuring burette is made at *l*. The filling of these double pipettes requires concise directions, and the reader is referred to Hempel-Dennis' "Gas Analysis" for these as well as for the general manipulation of the apparatus. When ready the tubes *k* and *e* and the bulb *a*, Fig. 369, are filled with the reagent, the spaces *b* to *f* with a gas free from oxygen, *c* and *g* with water, and *d* with air.

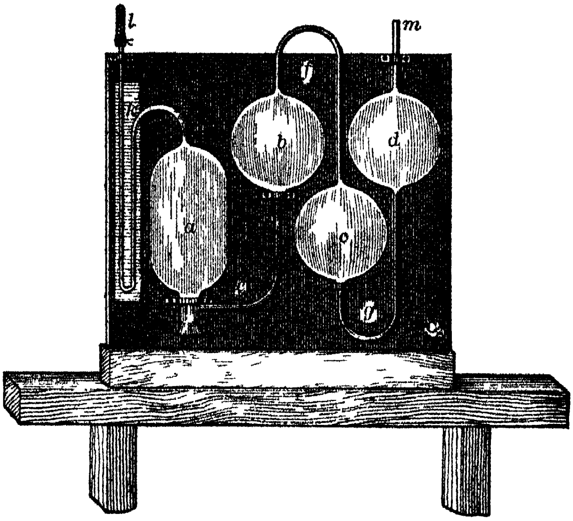

Fig 369 —Hempel Absorption Pipette, Double Form

282. Automatic Flue Gas Analysis Apparatus: CO_2 Recorders.— It has of late years become recognized that the efficiency of a furnace is largely a function of the percentage of CO_2 carried by the flue gases. This has led to the invention of a number of CO_2 indicators and recorders designed to keep a check upon the percentage of this gas. So far none of the apparatus developed has successfully attempted more than the determination of CO_2.

There are two fundamental types of this apparatus: the indicating and the recording. A type example of each will be given, although there are a number of others on the market.

Fig. 370 shows the Arndt econometer, which illustrates the indicating type. The gas from the boiler flue enters at the arrow marked accordingly and passes first an excelsior filter. After this it is passed through a cotton filter and then through a calcium chloride tube to dry it. The indicating apparatus consists of a very delicate balance mounted in an iron case. The left arm of the balance supports a glass bulb, while the other carries a small scale pan in which fine shot is placed until the pointer shows zero on the scale, when no gas is going through the apparatus. Up into the bulb there is passed a slender glass bulb perforated at the top. The two bulbs are independent of each other and do not touch. The course of the gas as it rises in the small tube and escapes into the larger one is shown by arrows. From here it is drawn out by an ejector or other means through the horizontal tube shown at the bottom. CO_2 is heavier than O_2; consequently, when the composition of the gas in the large bulb changes from that of air to that of flue gas, the bulb gains weight by an amount proportional to the increase in CO_2. The pointer then moves over the scale to the right, showing a certain amount of CO_2.

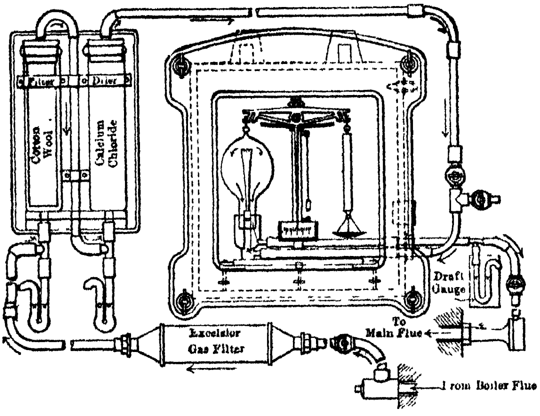

FIG. 370.—ARNDT ECONOMETER.

A type of the second class is the Sarco CO_2 recorder, Fig. 371.

This consists of an aspirator *Q* operated by water, which draws the gas into the apparatus and normally through pipes *D*, *C*, *E*. The mixture of water and gas is discharged into the vessel *L*, where a large part of the water is wasted through the overflow *R*. The rest flows through the tube *H* (the rate being adjusted by means of the cock *S*) into the vessel *K*. The construction of *K* is shown in section above. The water fills the upper compartment in *K*, tending to force the air in the latter through a connecting pipe into the lower compartment. This compartment is partly filled with a mixture of 1 part of glycerine to 2 parts of water. The air pressure generated forces the fluid out of the lower compartment, causing it to rise into the tubes *C* and *D*. As soon as the lower openings in *C* and *D* are closed off, gas can no longer pass through tube *E*, and for the time being the aspirator draws the gas past the seal *F*. A part of the gas sample trapped in the apparatus is forced out through the central tube just above *C*, against the slight resistance of the elastic bag *P*, by the rising liquid, but at the instant the latter reaches the mouth of the central tube the gas is completely confined. The design is such that just 100 c.c. of the gas are retained each time. The further rise of the liquid in *C* next forces the gas through *Z* over into the reagent bulb *A*, filled with KOH solution, where the CO_2 is absorbed. The caustic solution is forced over into *B*. The upper end of *B* is closed by a bell *N*, which floats in a glycerine seal. Central in *B* there is located a very fine tube which maintains communication with the atmosphere as long as the caustic solution has not risen far enough to seal its lower end. At the instant this happens, the air trapped in *B* will start to raise the bell *N*. The latter operates the pencil gear *XMY*, drawing a vertical line on paper, which is moved by

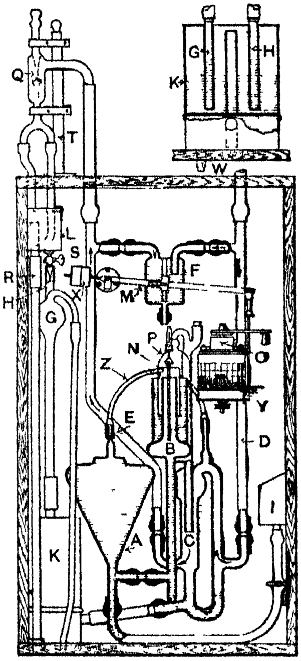

Fig. 371.—Sarco CO_2 Recorder.

the clockwork *O*. In every case the pencil will not start to operate until the lower end of the central tube in *B* is sealed. The amount of rise of the bell after that depends upon the amount of gas absorbed in *A*. If a large amount of CO_2 be present, the bell will not rise so far and the pencil line will be short; if the amount of CO_2 be less, the pencil will travel farther. In any case the travel is proportional to the amount of CO_2 absorbed. While this operation is going on in the treating bulb, the upper compartment of *K* has been filled with water and the latter starts to rise in *G*. This tube is a siphon which starts to operate as soon as the water reaches the top and will empty *K* very rapidly, allowing the displacement liquid to return to the lower compartment of *K*. A new sample of gas is then drawn in.

283. Assumptions and Accuracy of Flue Gas Analyses. — The analysis is called volumetric, although it is practically based upon the determination of partial pressures. When a number of gases are found together in a limited space, they are each uniformly distributed in that space. Each gas occupies the total volume at its own particular partial pressure. The total pressure exerted is the sum of all of the partial pressures. When one of the gases is absorbed, as in the analyses, and the remaining gases are compressed until the total pressure is again the same as it was before the absorption, a volume decrease is found; and, provided the temperature has been maintained constant and that the gases follow, with sufficient closeness, the law pv = constant for isothermal compression, the volume decrease observed will correctly represent the amount of gas which was absorbed. Within the degree of accuracy of other factors concerned in engineering gas analysis, the conditions above stated may be easily met, for the gases follow the law stated above closely enough and the temperature may be kept practically constant by surrounding the measuring tube of the apparatus with a water jacket.

A slight *theoretical error* is introduced into the analysis where the measurement of the change of volumes is made over water, on account of the fact that the gases will be saturated with water vapor. It can, however, be shown that the error thus made is negligible in flue gas analysis.

The *real errors* in the analysis are the following:

(1) Errors in getting a truly average sample.

(2) Failure to saturate the liquid in the sampling bottles and apparatus to equilibrium with the gas sample.

(3) Using reagents too long, so that they become weak and do not give complete absorption in reasonable time.

(4) Bad design of apparatus, not giving sufficient contact between reagent and gas.

(5) Attempts to force the rapidity of analysis.

Of the above items, (1) and (4) can be reduced to a minimum by sufficient care, although with regard to (4) the claim is made that complete CO absorption will not take place when the gases are simply in contact with the reagent and that shaking is necessary. Under (5) it should be noted that the gas should be driven over to the reagent tubes at least twice for every absorption, partly at least to account for the gas released from solution in the water of the measuring tube and partly to make sure that the absorption of the particular gas is complete.

Errors in the application of the flue gas analysis to engineering calculations also arise from the making of incomplete analyses. As already stated, there is a prevalent belief that it is sufficient to determine simply CO_2 and O_2. It is perhaps true that this is sufficient in the majority of cases. It is nevertheless advocated that the CO determination be also made in every case. The H_2 determination is of less importance. With respect to the CO determination, care should be taken to see that the O_2 absorption is complete, for the CO reagent will take up O_2, which will be charged up as CO if the O_2 was not previously completely removed.

284. Losses in Commercial Combustion. The Computation of the Flue Loss. — The losses in combustion taking place in the ordinary furnace may be tabulated as follows:

(1) Incomplete combustion
  (*a*) due to unconsumed fuel rejected with the ashes in the refuse,
  (*b*) due to unconsumed C, CO, H_2, and C_nH_{2n} in the flue gases.
    Unconsumed C forms *smoke.*

(2) Losses in sensible heat of the flue gases, apart from the loss in moisture.

(3) Loss due to moisture in the flue gases.

(4) Radiation and convection loss from the furnace.

(5) Radiation and convection loss from the boiler or other heat consumer.

These losses will be discussed at greater length in what follows, and methods of computing them will be developed where possible.

Loss under (1*a*). Unconsumed fuel in the refuse can be found from the weight of fuel fired and the *true ash* content of the fuel as determined in the laboratory. In general, the excess weight of the refuse over what should be the weight of true ash is not identical in quality with the original fuel fired. With coal, the excess weight is practically carbon, all volatile matter having been expelled by the high temperature. There are exceptions to this general rule in that in some cases a grate of bad design may waste a considerable part of the original fuel. In such a case it may be necessary to determine the actual heat content of the refuse in a calorimeter to accurately find the heat loss due to fuel in refuse. In accurate efficiency computations it may be best to use that method in any case. But where the accuracy of the refuse determination can be depended upon, the computation of the heat loss may be made according to the following formula:

$$\text{Heat loss due to fuel in refuse per pound of fuel fired}$$
$$= 14{,}540 \text{ (per cent of refuse} - \text{per cent of true ash) B.t.u.}$$

Losses under (1*b*), (2), *and* (3). The loss due to carbon (smoke) in the flue gases is in itself negligible even for the densest smoke, although far more evident to the eye than the other losses. Smoke results from the improper management of the combustion of the hydrocarbons. The presence of smoke means either (*a*) insufficient supply of air at the point where the hydrocarbons (volatile matter of coal, for instance) are burning, or (*b*) a temperature below the kindling temperature at the point where the air and the hydrocarbons meet and mix. Smokeless combustion therefore requires (*a*) a sufficient supply of air *above* the fuel bed to take care of the volatile hydrocarbons, and (*b*) sufficient size and proper arrangement of combustion chamber so that the combustion of the hydrocarbons may be complete before the products of combustion meet with any cooling surface, such as boiler tubes. Too great a supply of air above the fuel bed will, however, reduce the temperature of the products of combustion below the kindling point. Smokeless combustion is consequently a nice problem of adjustment of design and operation of the furnace with respect to the qualities of the fuel.

Smoke is an indicator of incomplete combustion, but the amount of heat lost is given very closely by the amounts of CO and H_2 found in the flue gas. Smoke measurement (see Article 287) in itself is of no definite value. In any case, its value can only be relative, and the same amount of smoke from different furnaces does not correspond to the same heat losses in the gas.

The computation of the heat losses under (1*b*), (2), and (3) requires a knowledge of the amount of the flue gases produced. This can be found from the amount and analysis of the fuel, the amount and analysis of the refuse, and the analysis of the flue gases. In the following computations it will be assumed that the fuel is coal. For the time being the water vapor present in the flue gas originating from the "moisture" in the coal and from "humidity" in the air used for combustion will be neglected, and only the vapor resulting from the actual combustion of hydrogen in the fuel will be considered.

The analysis as made of the flue gas does not check with the actual composition of the gas in the flue on account of changes made in the water vapor content inherent in our method of collecting and analyzing the gas. The gas in the flue carries an amount of water vapor which may or may not be sufficient to saturate the gas at flue temperature. Collecting the gas in the ordinary way is likely to affect the water vapor content and again the gas may or may not be saturated at the end of the collecting operation, depending, however, largely upon whether mercury or water is used as the displacement liquid. In the analysis, however, the gas comes in contact with water and *the assumption is fair that the gas is saturated with water vapor while being analyzed.* This means that as CO_2 is absorbed a certain proportional quantity of water vapor will also disappear and be accounted for as CO_2, and similarly for the O_2 and CO absorptions. It was stated in Article 283, however, that the error thus made is insignificant and *to all intents and pur-*

poses we may assume that the results of the flue gas analysis as ordinarily made are those for dry gas.

Of course the combustion of hydrogen in the furnace required a certain amount of oxygen out of the total contained in the air supplied. From the statements above made it will be clear that the flue gas analysis does not account for this, and is, therefore, not a sufficient basis upon which to compute flue gas volumes or weights per unit weight of fuel.

To obtain losses under (1 *b*), (2), and (3) the simplest way is to immediately convert the volume relations as given by the flue gas analysis for the *dry gas* to weight relations.

Let O_2, CO_2, CO, H_2, and N_2 represent the volume percentages (expressed as whole numbers, see example following) of the corresponding gases as found by the analysis. To reduce to a weight basis, multiply each one of these percentages by the corresponding density. This would not be a simple operation, since the densities are a function of temperature, if it were not for the fact that we can at once get to a *relative* weight basis by simply multiplying the volume percentage of each gas by the molecular weight of the gas, since densities of all perfect gases must be in proportion to their molecular weight.

The *relative weights* concerned are then:

$$32\ O_2,\ 44\ CO_2,\ 28\ CO,\ 2\ H_2,\ \text{and}\ 28\ N_2.$$

Now the chemical analysis of the coal shows how much carbon and hydrogen are available per pound of fuel fired. The carbon is not all burned on account of the loss in refuse (neglecting smoke); the hydrogen may not be all burned, in which case H_2 will appear in the flue gases.

Let *c* represent the weight of carbon actually burned (after deducting the loss in refuse) per pound of fuel fired. From the combustion formulas

$$1\ \text{lb.}\ C_2 + 2.66\ \text{lbs.}\ O_2 = 3.66\ \text{lbs.}\ CO_2$$

and

$$1\ \text{lb.}\ C_2 + 1.33\ \text{lbs.}\ O_2 = 2.33\ \text{lbs.}\ CO$$

it will be seen that 1 pound of CO_2 carries 1/3.66 pounds C_2, and that 1 pound CO carries 1/2.33 pounds C_2.

The weight c of carbon burned may then be accounted for by the relation

$$c = n\left(\frac{44\,CO_2}{3.66} + \frac{28\,CO}{2.33}\right),$$

where 44 CO_2 and 28 CO represent the relative weights of these particular gases in the flue gas and n is a factor to be determined.

Having found n, the real weights of the various gases *per pound of fuel* fired may then be found from the relations:

Weight of free oxygen $= n \cdot 32\ O_2$ pounds.
Weight of carbon dioxide $= n \cdot 44\ CO_2$ pounds.
Weight of carbon monoxide $= n \cdot 28$ CO pounds.
Weight of nitrogen $= n \cdot 28\ N_2$ pounds.

The weight of available hydrogen *not* burned is similarly equal to $n \cdot 2\ H_2$. Subtract this from the total hydrogen shown as available by the chemical analysis of the fuel and let h be the weight of hydrogen actually burned per pound of fuel. Then combustion will produce a weight of water vapor equal to $9\ h$ pounds.

The other two sources of water vapor may next be accounted for. The weight resulting per pound of fuel from the moisture in the fuel is given by the chemical or the proximate analysis. The third source is from humidity in air. This can be determined as follows: The weight of nitrogen above computed is practically that supplied by the total air used. Hence, since nitrogen is 77 per cent by weight of air, the total air supplied is $\frac{n \cdot 28\ N_2}{.77}$ pounds per pound of fuel. The water vapor carried in by the air per pound of fuel can next be found from hygrometer readings and humidity tables. See Chap. XXIII.

This determines the weight of all the individual gases making up the composite flue gas resulting from the combustion of one pound of the fuel under given conditions. We are next ready to compute the *heat losses* under (1*b*), (2), and (3).

(1*b*) Incomplete combustion of carbon (C_2) (smoke) may be neglected, as stated.

Incomplete combustion of carbon monoxide (CO). Each pound of CO that might have been burned to CO_2 represents a loss of

4380 B.t.u. Hence the loss per pound of coal is 4380 ($n \cdot 28$ CO) B.t.u.

Incomplete combustion of hydrogen (H_2). Each pound of H_2 in the flue gas represents a net loss of $(52{,}150 - 4.09\ t_f + 9\ t_r)$* B.t.u., where t_f equals temperature of flue. Hence the loss per pound of coal is $(52{,}150 - 4.09\ t_f + 9\ t_r)$ ($n \cdot 2\ H_2$) B.t.u.

(2) Losses in sensible heat. For oxygen, nitrogen, carbon dioxide, carbon monoxide, and hydrogen, the loss in sensible heat is in every case

$$(\text{Wt. of gas per lb. of coal}) \times (\text{mean sp. heat}) \times (t_f - t_r),$$

where t_f = flue temp., and t_r = room temp. For specific heats see Art. 253, p. 468, or Chap. XXI.

(3) The losses of heat due to moisture in the flue gases depend upon the source of the moisture. Each pound of moisture formed by the combustion of hydrogen and each pound brought in as "moisture" in coal carries with it

$$(1090.7 + .455\ t_f - t_r) \text{ B.t.u.}†$$

The moisture which enters the furnace as "humidity" in air carries with it only the heat required to superheat it to the temperature t_f, amounting to approximately $.46\ (t_f - t_r)$ per pound of moisture.

Losses under (4) *and* (5). These losses are rarely susceptible of direct determination and are usually determined together by difference between the other uses and losses of heat that can be accounted for and the total heat per pound of the fuel.

Example. — The coal analyses as follows:

C, 76 per cent; H, 5 per cent; O + N + S, 9 per cent; ash, 6 per cent; H_2O, 4 per cent.

Heating value, 14,000 B.t.u. per pound.

Ash determined on test, 8 per cent of coal fired.

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* This expression is derived as follows:

The loss per pound of H_2 not burned would be 61,950 B.t.u.

If this hydrogen had burned to water vapor, the loss would have been $9\ (1090.7 + .455\ t_f - t_r)$ B.t.u. per pound of hydrogen, which represents the heat carried away in water vapor.

The *net* loss per pound of H_2 therefore is:

$$61{,}950 - 9\ (1090.7 + .455\ t_f - t_r) = (52{,}150 - 4.09\ t_f + 9\ t_r) \text{ B.t.u.}$$

† See Art. 252.

Temperature of flue, 525°; room, 72°.

Humidity of air, 60 per cent.

Flue gas analysis, O_2, 10 per cent; CO_2, 8 per cent; CO, .5 per cent; N_2 (by difference), 81.5 per cent; H_2 not determined.

(1) Relative weights:

For $O_2 = 32 \times 10$; $CO_2 = 44 \times 8$; $CO = 28 \times .5$; $N_2 = 28 \times 81.5$.

(2) Carbon burned:

Carbon lost in refuse = .08 − .06 = .02 pound per pound of coal.

Carbon burned = .76 − .02 = .74 pound per pound of coal.

(3) Value of n:

$$.74 = n\left(\frac{44 \times 8}{3.66} + \frac{28 \times .5}{2.33}\right).$$

$$n = .0072.$$

(4) Weight of the various products of combustion except moisture per pound of coal:

Oxygen = $.0072 \times 32 \times 10 = 2.31$ pounds.

Carbon dioxide = $.0072 \times 44 \times 8 = 2.54$ pounds.

Carbon monoxide = $.0072 \times 28 \times .5 = .10$ pound.

Nitrogen = $.0072 \times 28 \times 81.5 = 16.40$ pounds.

Unburned hydrogen = assumed zero.

(5) Weights of water vapor per pound of coal:

(*a*) From the combustion of hydrogen in the coal = $9 \times .05 = .45$ lb.

(*b*) From moisture in the coal = .04 lb.

(*c*) Brought in by air:

Air supplied = $\dfrac{16.40}{.77} = 21.3$ lbs.

At temperature of 72 degrees and humidity of 60 per cent, weight of water vapor carried = .01 pound per pound of air. Hence water vapor as humidity = .21 pound.

(6) Heat losses:

(*a*) From carbon in refuse = 14,540 (.08 − .06) = 292 B.t.u.

(*b*) Due to incomplete combustion of CO = $4380 \times .10 = 438$ B.t.u.

(*c*) Due to sensible heat in:

$O_2 = 2.31 \times .220 \times (525 - 72) = 231$ B.t.u.

$CO_2 = 2.54 \times .222 \times (525 - 72) = 256$ B.t.u.

$CO = .10 \times .240 \times (525 - 72) = 11$ B.t.u.

$N_2 = 16.40 \times .249 \times (525 - 72) = 1849$ B.t.u.

2347 B.t.u.

(*d*) Due to water vapor:

From combustion of hydrogen and moisture in fuel

$= (.45 + .04)\ (1090.7 + .455 \times 525 - 72) = 616$ B.t.u.

In humidity = $.21 \times .46\ (525 - 72) = 44$ B.t.u.

(7) Percentage of losses:

Heating value of coal = 14,000 B.t.u. per pound.

	B.t.u.	Per cent.
In refuse	292	2.08
Incomplete combustion	438	3.13
Sensible heat of gases	2347	16.78
Heat in water vapor	660	4.72

Total loss in flue gas alone = 3.13 + 16.78 + 4.72 = 24.63 per cent of the heat in the coal.

285. Errors in the Flue Gas Calculations as Above Carried Out.—

(1) Unburned hydrogen in the gases was neglected. This can usually be done without sensible error, as the occurrence of H_2 in the flue gases in any considerable quantity is not likely.

(2) Loss of carbon in smoke was neglected. This also is a negligible error.

(3) Sulphur in the coal was neglected. A close enough correction for this is to use $(c + s)$ in place of c in the formulas above. SO_2 will be indicated as CO_2 in the analysis.

(4) Nitrogen in the coal was neglected. No appreciable error can arise from this, as the amount of nitrogen supplied in the air is so much greater in amount.

(5) Oxygen in the coal was neglected. It may exist as oxygen of nitrates, carbonates, or carbohydrates. The oxygen of nitrates and carbonates is all that will be found in anthracites and most bituminous coals. It is not available for combustion. N_2O_5 from the nitrates will occur in the flue gases, but it will be absorbed in the water of the sampling bottles and will not show in the analyses. CO_2 from carbonates will be shown in the gas analysis. The amount of CO_2 from this source is almost always negligibly small as compared with the amount from combustion of carbon. With lignites the oxygen from carbohydrates may be said, in a way, to be available for combustion; but it will be small in amount as compared with the oxygen supplied in air.

286. The Excess or Dilution Coefficient. — This coefficient is usually defined as the ratio of the weight or volume of air which was actually supplied to a furnace for the combustion of one pound of the fuel as fired to the weight or volume of air which is in theory required to burn one pound of the fuel completely. The ratio is often used in flue gas computations and several formulas are usually given for it. They are sometimes seriously in error in that they neglect the oxygen bound by the combustion of hydrogen in the fuel which is not accounted for in the flue gas analysis, as was above pointed out.

The simplest method of deriving the formula for this coefficient is as follows:

In the flue gas analysis, O_2 (free oxygen) is evidently the oxygen brought into the furnace by the excess air. The nitrogen which was a part of this excess air is, then, $\frac{79}{21}\,O_2$

Now the N_2 shown by the analysis is practically all due to the total air entering the furnace, hence $N_2 - \frac{79}{21}\,O_2$ represents the nitrogen which was a part of the air that was used for combustion. From the definition of excess coefficient we will then have

$$X = \frac{N_2}{N_2 - \frac{79}{21}\,O_2}, \tag{I}$$

since in each case the amount of nitrogen can serve as a measure of the amount of air of which it was a part.

Form (I) is in very common use, but must be modified as follows as soon as the gas contains CO.

$$X = \frac{N_2}{N_2 - \frac{79}{21}\left(O - \frac{CO}{2}\right)}. \tag{II}$$

Still another form given by some writers is derived as follows:

Let L' = volume of air supplied to furnace per pound of fuel.

Let L'' = volume of air theoretically needed for complete combustion of one pound of the fuel.

Then $$X = \frac{L'}{L''} = \frac{O_2' + N_2'}{O_2'' + N_2''} = \frac{O_2' + \frac{79}{21}\,O_2'}{O_2'' + \frac{79}{21}\,O_2''} = \frac{O_2'}{O_2''}.$$

Now if, as before, O_2 represents the free oxygen found by the analysis, then

$$O_2 = O_2' - O_2'', \text{ or } O_2'' = O_2' - O_2.$$

Hence $$X = \frac{O_2'}{O_2' - O_2}$$

$$= \frac{21}{21 - O_2}, \tag{III}$$

since the oxygen in the original air must have been 21 per cent by volume.

To compare these formulas with one another, take the gas analysis of the previous example. The results will be:

$$\text{Formula I,} \qquad X = \frac{81.5}{81.5 - \frac{79}{21}.10} = 1.86.$$

$$\text{Formula II,} \qquad X = \frac{81.5}{81.5 - \frac{79}{21}\left(10 - \frac{.5}{2}\right)} = 1.81.$$

$$\text{Formula III,} \qquad X = \frac{21}{21 - 10} = 1.90.$$

The air required in theory by the coal of the above example is:

$$\text{For carbon} = (.76 - .02)\frac{2.66}{.23} = 8.57 \text{ lbs.}$$

$$\text{For hydrogen} = .05 \times \frac{8}{.23} = 1.74 \text{ lbs.}$$

$$\text{Total,} \qquad \overline{10.31} \text{ lbs.}$$

It was computed under (5), p. 537 above, that the air supplied per pound of fuel was 21.3 pounds. Hence, the true excess coefficient is

$$X = \frac{21.3}{10.31} = 2.07.$$

This result shows how the neglect to account for oxygen used in combustion of hydrogen may seriously affect the heat loss computation in flue gas if the approximate formulas are used. Of course the smaller the percentage of hydrogen in the fuel, the smaller the discrepancy, and for some hard coals the difference may be negligible.

In flue gas computations the coefficient X is used to determine the extra air supplied after the theoretical amount of air necessary per pound of fuel is found by computation based in the usual way on the chemical analysis of the coal. It is recommended, however, that the method of computing outlined in Article 284 be used because it is of general applicability and is not subject to error concerning the amount of oxygen used in the combustion of hydrogen.

287. Smoke Determinations. — These may be either *quantitative* or *relative.*

Quantitative determinations are not often made, as the method is too elaborate for ordinary testing. The actual weight of unburned C in the flue gases is found by drawing through a fine filter, which retains all of the finely divided carbon, a volume of gas which must be measured. The amount of carbon collected is then usually found by some combustion process in the chemical laboratory.

There are a number of methods of determining smoke relatively. The one probably most used in this country is that devised by Ringelmann. This method uses the fact that if a network of black lines of varying thickness but equal distances apart is drawn on a white ground, the chart will assume to the eye an even gray tone when placed at a distance of 45–60 feet from the observer. The depth of the color depends upon the thickness of the lines, and this scheme not only gives a method of obtaining a series of gray colors but also allows of their exact duplication in different localities, thus establishing in a sense a standard.

To construct the Ringelmann chart, take a strip of white cardboard and lay out on it six squares in a row, each 100 × 100 millimeters. Divide each square, except the one at the left, into smaller squares. The thickness of the cross lines in squares I to IV inclusive should be according to the schedule following, square V being solid black:

Square or Color Number.	Thickness of Line, mm.	Distance in the Clear between Lines, mm.
0	No lines, clear white field	....
I	1.0	9.0
II	2.3	7.7
III	3.7	6.3
IV	5.5	4.5
V	Solid black	....

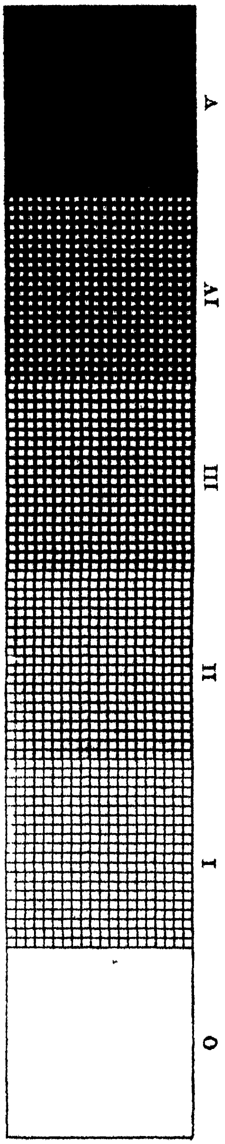

FIG. 372. RINGELMANN SMOKE CHART.

During a test, place this chart at a distance of 45·60 feet from the observer in such a manner that the latter can also observe the smoke at the same time. Take readings, say every five minutes, and continue for one minute, taking as many readings as possible in that time. Estimate with which one of the color series the smoke matches and record the number. Use the average of the numbers as the smoke number for that particular minute.

It must be apparent that the results obtained by this method will depend very largely upon the observer, and it is quite likely that any two observers, working independently, will get very different results. This has been recognized and several instruments have been devised to overcome the difficulties involved. Among them may be mentioned the Dosch smoke observer, Fig. 372a. This consists of a tube AB with a branch tube R_2. D is a circular piece of glass, pivoted at E, which has been divided into a number of sectors, usually six. The sectors are colored, varying from white to black. F is a source of illumination with a reflector G. The light coming from D is thrown upon the mirror C, which at the center has a circular opening O. The end A of the tube AB having been placed to the eye and the tube directed at the smoke to be observed, E is turned until the light coming from D, and reflected in C, blends with the color in the opening O. The number can then be read directly by noting the position of D. This instrument gives quick readings and is much more reliable than the Ringelmann chart.

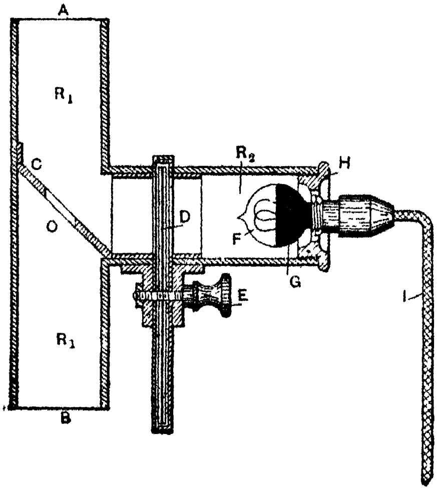

Fig. 372 A.—Dosch Smoke Observer

CHAPTER XIV.

METHODS OF DETERMINING THE AMOUNT OF MOISTURE IN STEAM.

288. Meaning of the Term "Quality of Steam." — Steam at any given pressure may show any one of three conditions of quality: it may be wet saturated, dry saturated, or superheated. The quantity of heat contained in each pound of steam will vary with each change in quality at the given pressure. The heat of the liquid, q (see Art. 179), is common to the steam under all three conditions. Next, if the liquid is completely converted into vapor at the given pressure, it will render latent the quantity of heat r (see Art. 179), and the resulting steam is said to have a quality equal to 1.0. The steam is then said to be in the *dry saturated condition.* Assuming, however, that not all of the liquid be converted into vapor, but that at the end of the process only x per cent of the liquid has been vaporized, then the total heat in unit weight of such steam will be

$$= (xr + q) \text{ B.t.u.},$$

and the steam is said to be in the *wet saturated condition,* often called simply *wet steam.* The percentage x is then said to be the quality of steam, and it expresses practically the ratio between the latent heat that the sample of steam actually contains and the latent heat that it should contain if it were dry saturated. The fraction of water remaining $(1 - x)$ is assumed to be mechanically distributed in the dry steam and carried along with it. To attain the superheated condition, the heat supply to the steam is not interrupted after all the liquid is vaporized (see Art. 179). The result is an increase in the temperature of the steam, and if the heating is done at constant pressure (usual case) and the temperature of the steam at the end of the process is $T_2°$, while the saturation temperature for the pressure is $T_1°$, so that $T_2 - T_1 = D$, the total heat in steam will then be

$$= (r + q + C_pD) \text{ B.t.u.},$$

in which C_p is the mean specific heat in the range T_1 to T_2.

Now from the above definitions of quality for wet and dry saturated steam it appears that quality may be expressed as the quotient of the heat remaining in a given weight of steam after the heat of the liquid is subtracted from the total, divided by the latent heat r at the given pressure. Following out the analogy in the case of *superheated steam*, we then will have

$$\text{Quality } x = \frac{r + C_p D}{r}.$$

This expression is of course always greater than 1.0. The result, however, is meaningless unless the pressure is stated at the same time, and in commercial practice it is usual to transpose it to degrees of superheat D, thus:

$$D = \frac{r(x - 1)}{C_p}.$$

289. Importance of Quality Determinations.—The importance of correctly determining the quality of steam is great, because the percentage of water carried over in the steam in the form of vapor or drops of water may be large, and this water is an inert quantity so far as its power of doing work is concerned, even if not a positive detriment to the engine. Any tests for the efficiency of engine or boiler not accompanied with determinations of the amount of water carried over in the steam would be defective in essential particulars, and might lead to erroneous results.

On the other hand, the greatest precautions should always be taken to insure an average sample of steam for quality determination. Carelessness in this respect may easily lead to errors larger than would be the case if the quality determination had been omitted altogether.

290. Methods of Determining the Quality. — The methods of measuring the amount of moisture contained in steam may be considered under three heads: first, *Calorimetry proper*, in which the method is based on some process of comparing the heat actually existing in a pound of the sample with that known to exist in a pound of dry and saturated steam at the same pressure; second, *Mechanical Separation* of the water from the steam, involving the processes of separation and of weighing; third, *Chemical Methods*, in which case a soluble salt is introduced into the water of the boiler.

All methods for determining the quality of steam are included under the head of *calorimetry*, and instruments for determining the quality are termed *calorimeters.*

291. Classification of Calorimeters. — The following classification of different forms of calorimeters is convenient and comprehensive:

- Calorimeters
  - Condensing
    - Jet
      - Barrel or Tank.
      - Continuous.
    - Surface
      - Hoadley Calorimeter.
      - Barrus — Continuous.
  - Superheating
    - External — Barrus Superheating.
    - Internal — Peabody Throttling.
    - Electric Superheating — Thomas.
  - Directly determining moisture
    - Separator.
    - Chemical.
  - Universal or Combination
    - Ellyson.

In what follows, the construction and method of use of a representative type of each kind of calorimeter will be briefly discussed, and errors will be pointed out wherever their possible occurrence is likely to affect the results seriously.

292. Use of Steam-tables. — In reducing calorimetric experiments steam-tables will be required. The explanation of the terms used will be found in Article 180, page 335, and tables will be found in the Appendix of this book.

Students will please notice that the pressures referred to in the steam-tables are *absolute* not *gauge* pressures, and that gauge pressures must be reduced to absolute pressures by adding the barometer reading reduced to pounds per square inch, before using the tables.

The following symbols will be employed to represent the different properties of steam:

TABLE OF SYMBOLS.

Properties of Steam.	Symbol.	Properties of Steam	Symbol.
Pressure, pounds per sq. in.	p	Total heat, B.t.u.	λ or H
Pressure, pounds per sq. ft.	P	Weight of cu. ft. of steam, lbs.	δ
Temperature, degrees Fahr.	t	Vol. of 1 lb. steam, cu. ft.	v
Temperature, absolute	T	Vol. of 1 lb. water, cu. ft.	σ
Heat of the liquid	q or S	Change in volume $v - \sigma$	u
Internal latent heat.	ρ or I	Quality of steam	x
External latent heat.	APu or E	Per cent of moisture	$1 - x$
Total latent heat	r or L	Degree of superheat	D

293. Method of Obtaining a Sample of Steam. — It has already been pointed out that the obtaining of a representative sample of steam is of the utmost importance in steam calorimetry. It is usually necessary to so arrange the apparatus that only a small percentage of the total steam passing can be examined in a calorimeter. Unfortunately there are certain practical conditions which make the taking of an average sample anything but certain.

From experiments made by the author, it is quite certain that the quality varies greatly in different portions of the same pipe, and that it differs more in horizontal than in vertical pipes. Steam drawn from the surface of the pipe is likely to contain more than the average amount of moisture; that from the center of the pipe to contain less. Perhaps the best method for obtaining a sample of steam is to cut a long-threaded nipple into which a series of holes is drilled, and screw this well into the pipe. Half-inch pipe is generally used for calorimeter connections, and it may be screwed into the main pipe one-half or three-quarters of the distance to the center, with the end left open and without side perforations, as shown in Fig. 373, or screwed three-quarters of the distance across the pipe, a series of holes drilled through the sides, and the end left open or stopped, as shown in Fig. 374. A lock nut on the nipple, which can be screwed against the pipe when the nipple is in place, will serve to make a tight joint. The best form of nipple is not definitely determined, although many experiments have been made for this purpose; a form extending nearly across the pipe and provided with a slit or with numerous holes is probably preferable. When the current of steam is ascending in a vertical pipe, the water seems to be more uniformly mixed than when descending in a vertical pipe or when moving in a horizontal one. There is, however, consider-

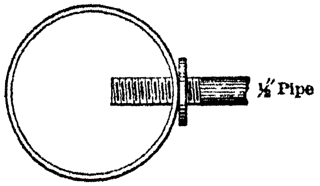

FIG. 373.

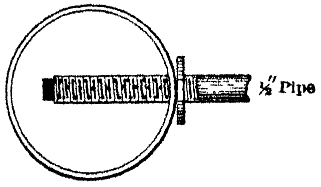

FIG. 374.

able variation for this condition, especially if the steam contains more than 3 per cent of water.

294. Method of Inserting Thermometers. — In the use of calorimeters it is frequently necessary to insert thermometers into the steam in order to correctly measure the temperature, a matter scarcely less important than the proper taking of the sample. For this purpose thermometers can be had mounted in a brass case, as shown in Fig. 375, which will screw into a threaded opening in the main pipe.

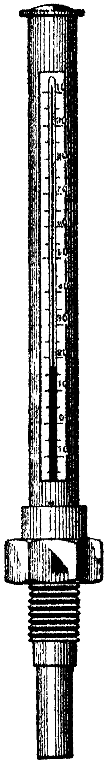

FIG. 375.

The author prefers to use instead a thermometer cup of the form shown in Fig. 376, which is screwed into a tapped opening in the pipe. Cylinder oil or mercury is then poured into the cup, and a thermometer inserted with graduations cut on the glass. The thermometer cups are usually made of a solid brass casting, the outside being turned down to the proper dimensions and threaded to fit a ¾-inch pipe fitting. The inside hole is drilled ½ inch in diameter, and the walls are left 1/16 inch thick. It should be noted that mercury cannot be used with a brass cup on account of amalgamation. In such cases iron or steel will have to be used. The total length of the cups may vary from 2 to 6 inches, depending on the place where they are to be used. In any case it is essential that the thermometer be inserted deep into the current of steam or water, and that no air pocket forms around the bulb of the thermometer. The thermometer should be nearly vertical, and as much of the stem as possible should be protected from radiating influence.

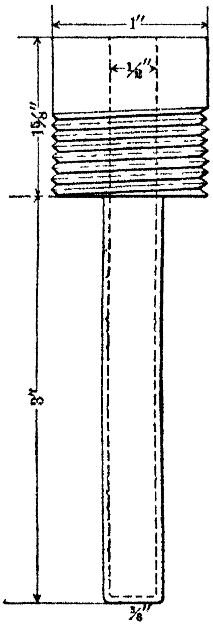

FIG. 376.—THERMOMETER CUP OR WELL.

If the thermometer is to be inserted into steam of very little pressure, the stem of the thermometer can be crowded into a hole cut in a rubber cork which fits the opening in the pipe. In case the thermometer cannot be inserted in the pipe it is sometimes bound on the outside, being well protected from radiation by hair felting; but this practice cannot be recommended, as the reading is often much less than is shown by a thermometer inserted in the current of flowing steam.

295. Condensing Calorimeters. — Condensing calorimeters are of two general classes: 1. The jet of steam is received by the condensing water, and the condensed steam intermingles directly with the condensing water. 2. The jet of steam is condensed in a coil or pipe arranged as in a surface condenser, and the condensed steam is maintained separate from the condensing water.

The principle of action of both of these calorimeters is essentially the same; the first class is, however, probably more often used than the second. The equations involved are as follows:

Let W = weight of condensing water.
w = weight of condensed steam.
t_1 = temperature of condensing water, cold.
t_2 = temperature of condensing water, warm.
t_3 = temperature of condensed steam.
K = water equivalent of calorimeter (see next article).

Now the heat gained by the calorimeter upon condensing one pound of steam will be equal to

$$\frac{W + K}{w}(t_2 - t_1) \text{ B.t.u.} \tag{1}$$

The heat in one pound of the sample steam above 32° F., as taken from the steam-table, for the pressure of the original sample, may be expressed by $xr + q$. The condensed steam is, however, not cooled to 32° F., but to some temperature t_3 usually higher than 32°. Then the net amount of heat per pound of steam condensed will be

$$xr + q - (t_3 - 32) \text{ B.t.u.} \tag{2}$$

The loss of heat by radiation must either be determined and added to equation (1) or it must be compensated for by so conducting the trial that t_1 is as much below the temperature of the room at the start as t_2 is above that temperature at the end of the test. Assuming that this method is used, we then have

$$\frac{W+K}{w}(t_2-t_1)=[xr+q-(t_3-32)], \qquad (3)$$

from which

$$x=\frac{\frac{W+K}{w}(t_2-t_1)-q+t_3-32}{r}. \qquad (4)$$

In the case of calorimeters of the first class, $t_3 = t_2$ and equation (4) becomes

$$x=\frac{\frac{W+K}{w}(t_2-t_1)-q+t_2-32}{r}. \qquad (5)$$

If x turns out to be greater than 1.0, compute the degree of superheat D from the equation

$$D=\frac{r(x-1)}{C_p}. \qquad (6)$$

296. Determination of the Water Equivalent of the Calorimeter.— The calorimeters exert some effect on the heating of the liquid contained in them, since the substance of the calorimeter must also be heated. This effect is best expressed by considering the calorimeter as equivalent to a certain number of pounds of water producing the same result. This number is termed the *water equivalent* of the calorimeter. The water equivalent, K, can be found in three ways*:

1. By computation from the known weight and specific heat of the materials composing the calorimeter. Thus, let C be the specific heat, W_c the weight; then

$$K = CW_c.$$

2. By drawing into the calorimeter, when it is cooled down to a low temperature, a weighed quantity of water of higher temperature and observing the resulting temperature. Thus, let W equal the weight of water, t_1 the first and t_2 the final temperatures,

---

* See also the discussion on the same subject in Chapter XIII, p. 478.

and K the water equivalent sought. Since the heat before and after this operation is the same,

$$(W+K)\,t_2 = Wt_1;$$

from which

$$K = \frac{W(t_1 - t_2)}{t_2}.$$

Radiation, however, affects the accuracy of the determination and must be corrected for.

3. By condensing a quantity of steam known to be dry saturated. Then, $x = 1.0$, and the equation of the previous article may then be used to determine K.

297. Forms of Condensing Calorimeters. — *Barrel or Tank Calorimeter.* — The barrel calorimeter belongs to that class of condensing

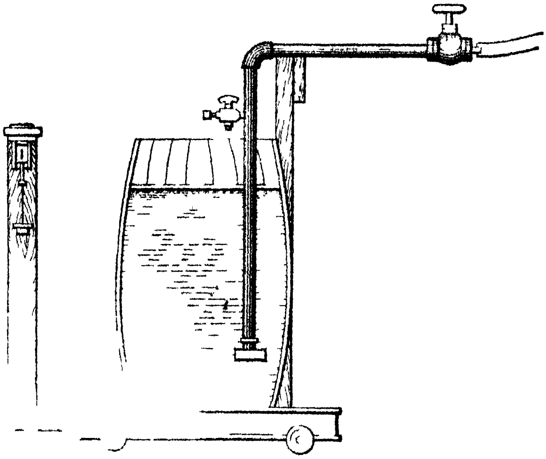

FIG. 377.—BARREL CALORIMETER

calorimeters in which a jet of steam intermingles directly with the water of condensation. It is made in various ways; in some instances the walls are made double and packed with a non-condensing substance, as down or hair felting, to prevent radiation, and the instrument is provided with an agitator consisting of paddles fastened to a vertical axis that can be revolved and the water thoroughly mixed; but it usually consists of an ordinary wooden tank or barrel resting on a pair of scales, as shown in Fig. 377.

A sample of steam is drawn from the main steam pipe by connections, as explained in Article 293, and conveyed by hose, or partly by iron pipe and partly by hose, to the calorimeter. In the use of the instrument, water is first admitted to the barrel and the weight accurately determined. The pipe is then heated by permitting steam to blow through it into the air; steam is then shut off, the end of the pipe is submerged in the water of the calorimeter, and steam turned on until the temperature of the condensing water is about 110° F. The pipe is then removed, the water vigorously stirred, the temperature and the final weight taken.

A tee screwed crosswise of the pipe, as shown in Fig. 377, forms an efficient agitator, provided the temperature be taken immediately after the steam is turned off.

The pipe may remain in the calorimeter during the final weighing if supported externally and if air be admitted so that it will not keep full of water; in such a case, however, it should also be in the barrel during the first weighing, or else the final weight must be corrected for displacement of water by the pipe. The effect of displacement is readily determined by weighing with and without the pipe in the water of the calorimeter.

The determination of the water equivalent of the barrel calorimeter will be found very difficult in practice, and it is usually customary to heat the barrel previous to using it, and then neglect any effect of the calorimeter. The accuracy of this instrument depends principally on the accuracy with which the temperature and the weight of the condensed steam are obtained. The conditions for obtaining the temperature of the water accurately are seldom favorable, as it is nearly impossible to secure a uniform mixture of the hot and cold water; the result is that determinations made with this instrument on the same quality of steam often vary from 3 to 6 per cent. From an extended use in comparison with more accurate calorimeters, the author would place the average error resulting from the use of the barrel calorimeter at from 2 to 4 per cent.

Example. — Temperature of condensing water, cold, t_1, is 52.8° F.; warm, t_2, 109.6° F. Steam pressure by gauge, 79.7; absolute, 94.4. Entering steam, normal temperature, from steam-table, t, 323.7° F. Latent heat, r, 891.2 B.t.u., q = 294 B.t.u. Weight of condensing water, cold, W, 360 pounds; warm,

$W + w$, 379.1 pounds; wet steam, w, 19.1 pounds. Calorimeter equivalent eliminated by heating. The quality

$$x = \frac{\frac{360}{19.1}(109.6 - 52.8) - 294 + 109.6 - 32}{891.2} = \frac{854.3}{891.2} = 95.9 \text{ per cent.}$$

298. Directions for Use of the Barrel Calorimeter. — *Apparatus.*— Thermometer reading to $\frac{1}{5}$ degree F., range 32° to 212°; scales reading to $\frac{1}{20}$ of a pound; barrel provided with means of filling with water and emptying; proper steam connections; steam gauge or thermometer in main steam pipe.

1. Calibrate all apparatus.

2. Fill barrel with 360 pounds of water, and heat to 130 degrees by steam; waste this and make no determinations for moisture. This is to warm up the barrel.

3. Empty the barrel, take its weight, add quickly 360 pounds of water, and take its temperature.

4. Remove steam pipe from barrel; blow steam through it to warm and dry it; hang on bracket so as not to be in contact with barrel; turn on steam, and leave it on until temperature of resulting water rises to 110° F. Turn off steam; open air cock at steam pipe.

5. Take the final weights with pipe in barrel, in same position as in previous weighings; also take weights with the pipe removed: calculate from this the displacement due to pipe, and correct for same.

Alternative for fourth and fifth operations. — Supply steam through a hose, which is removed as soon as water rises to a temperature of 110° F. Weigh with the hose removed from the barrel. Stir the water while taking temperatures.

6. Take five determinations, and compute results as explained. Fill out and file blank containing data and results.

7. Compute the value of the water equivalent, K, in pounds by comparing the different sets of observations.

299. The Continuous Jet Condensing Calorimeter. — A calorimeter may be made by condensing the jet of steam in a stream of water passing through a small injector or an equivalent instrument. The method is well shown in Fig. 378. A tank of cold water, B, placed upon the scales R, is connected to the small injector by the pipe *C*; the injector is supplied with steam by the pipe *S*, the pressure of which is taken by the gauge *P*; the temperature of the cold water is taken at *e*, that of the warm water at *g*. Water is discharged into the weighing tank *A*. The amount taken from the tank *B* is the weight of cold water *W*; the difference in the respective weights of the water in tank *A* and that taken from *B* is the weight of the steam *w*. The quality is computed exactly as for the barrel calorimeter.

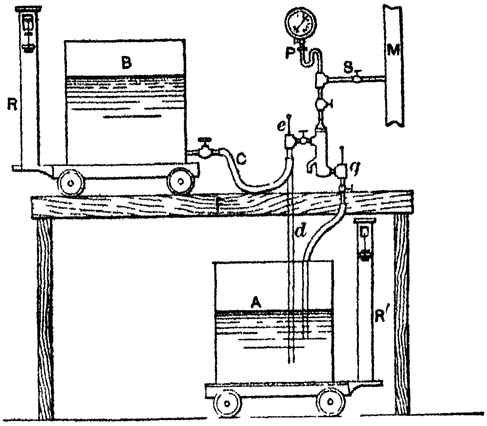

FIG. 378 — CONTINUOUS JET CONDENSING CALORIMETER

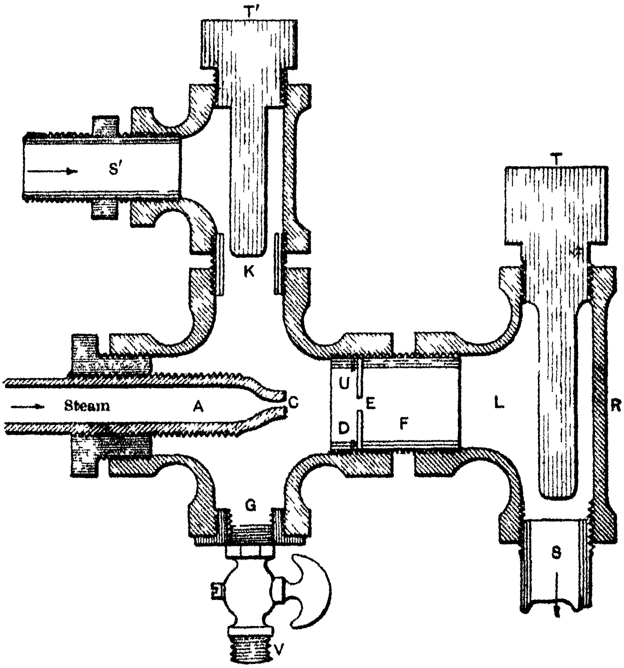

FIG. 379 — MIXING DEVICE FOR CONDENSING CALORIMETER.

In case a regular injector is used, as shown in Fig. 378, the tank *B* is not needed: water can be raised by suction from the tank *A* through the pipe *d*. The original weight of *A* will be that of the cold water; the final weight will be that of steam added to the cold water.

In case an injector is not available, and the water is supplied under a small head, a very satisfactory substitute can be made of pipe fittings, as shown in Fig. 379. In this case, steam of known pressure and temperature is supplied by the pipe *A*, cold water is received at *S'*, and the warm water is discharged at *S*. The temperature of the entering water is taken by a thermometer in the thermometer cup *T'*, that of the discharge by a thermometer at *T*. The steam is condensed in front of the nozzle *C*.

This class of instruments presents much better opportunities for measuring the temperatures accurately than the barrel calorimeter, and the results are somewhat more reliable.

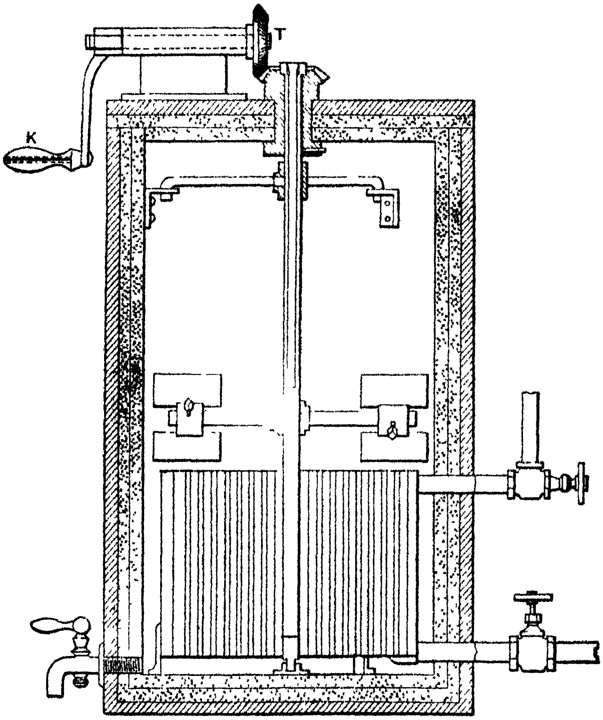

FIG. 380. — HOADLEY STEAM CALORIMETER.

300. The Hoadley Calorimeter. — This instrument belongs to the class of non-continuous surface calorimeters. It was described in

"Transactions of the American Society of Mechanical Engineers," Vol. VI, and consists of a condensing coil for the steam, situated in the bottom of a tank calorimeter, very carefully made to prevent radiation losses (Fig. 380). The dimensions were 17 inches diameter by 32 inches deep, with a capacity of about 200 pounds of water. The calorimeter was made of three concentric vessels of galvanized iron, the spaces being filled with hair felt and eiderdown. The condenser consisted of a drum through which passed a large number of half-inch copper tubes, the steam being on the outside, the water on the inside, of these tubes, the agitator consisting of a propeller wheel attached to an axis that could be rotated by turning the external crank *K*, effectually stirring the water. The thermometer for measuring the temperature was inserted in the axis of the agitator at *T*. In the hands of Mr. Hoadley the instrument gave accurate determinations.

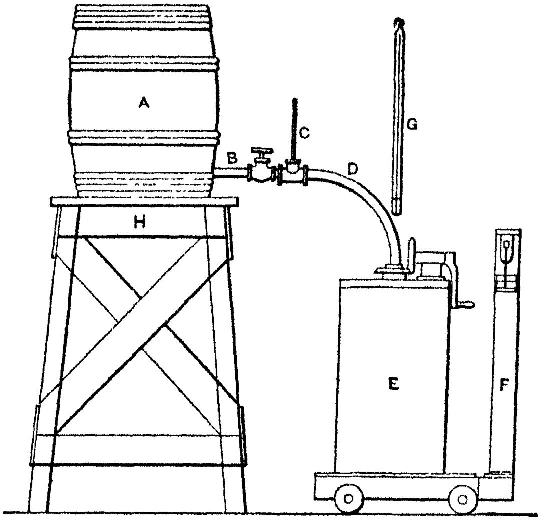

FIG. 381 — HOADLEY CALORIMETER.

In practice the instrument was arranged as in Fig. 381; the calorimeter *E* was placed on the scales *F* and supplied with cold water from the elevated barrel *A*. The temperature of the entering water was taken at *C*. Steam was admitted to the condensing coil until the temperature of the condensing water reached, say, 110° F. The weights before and after adding steam were taken by the scales *F*; the temperature of the warm condensing water was taken by a thermometer, *G*, inserted in the axis of the agitator. The water equivalent was determined as explained in article 296, and the quality computed by equation (4). The rate of cooling was determined, and an equivalent amount added as a correction for any loss of heat by radiation.

301. The Barrus Continuous Calorimeter. — This calorimeter is shown in Fig. 382. It consists of a steam pipe, *aJ*, surrounded by a tub or bucket, *O*, into which cold water flows; the condensing water is received as it enters the bucket in a brass tube, *k*, surrounding the pipe *a*, and is conveyed over and under baffle plates, *m*, so as to be thoroughly mixed with the water in the vessel, and is finally discharged at *c*. Thermometers are placed at *f* and at *g* to take the temperature of the water as it enters and leaves, and finally the condensing water is caught from the overflow and weighed. The condensed steam falls below the calorimeter; by means of the water-gauge glass at *e* it may be seen and kept at a constant height. The temperature of the condensed steam while it is still under pressure is shown by a thermometer at *h*. In order to use the calorimeter it is necessary to weigh the condensed steam; this cannot be done without further cooling, as it would be converted into steam were the pressure removed. For this purpose it is passed through a coil of pipe immersed in a bucket filled with water. The water

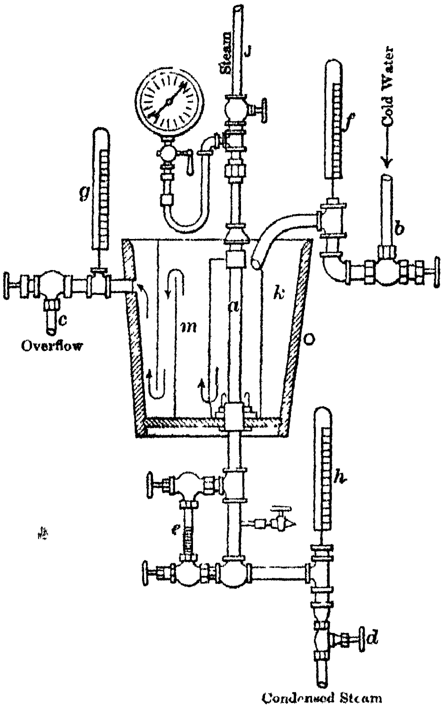

FIG. 382.—BARRUS CONTINUOUS CALORIMETER.

used in the cooling bucket has no effect on the quality of the steam and is not considered in the results; it is allowed to waste, but the condensed steam is caught and weighed.

The method of computing the quality of the steam from the observed data should be clear from previous articles.

The following form for recording readings and results obtained with this calorimeter will be found convenient:

SIBLEY COLLEGE—DEPARTMENT OF EXPERIMENTAL ENGINEERING.

DETERMINATION OF MOISTURE IN STEAM BY CONDENSING CALORIMETER.

MADE BY	Ithaca, N. Y.,....................191..
....	Type of Calorimeter..................
....	Barometer..................inches Hg.
....	Room Temp. ....

Duration of run — min...								
Pressures								
Gauge..............								
Absolute								
Weight of barrel								
Weight of water								
Tare								
Total.								
Weight of Steam........								
Tare								
Total.								
Temperatures..........								
Condensing water — initial.............								
Condensing water — final.....								
Steam — initial.								
Steam (condensed) — final..............								
Weight of water — lbs.								
Weight of steam — lbs.								
Water equivalent of Calorimeter.								
Ratio of water to steam.								
Latent heat of steam at initial pressure ..								
Quality — per cent....								
Moisture — per cent..								
Total heat in 1 lb. steam as determined ..								

302. Superheating Calorimeters. — These are of two forms, the external and the internal. The former is typified by the Barrus superheating calorimeter, the latter by the throttling calorimeter, of which the Peabody is the original example.

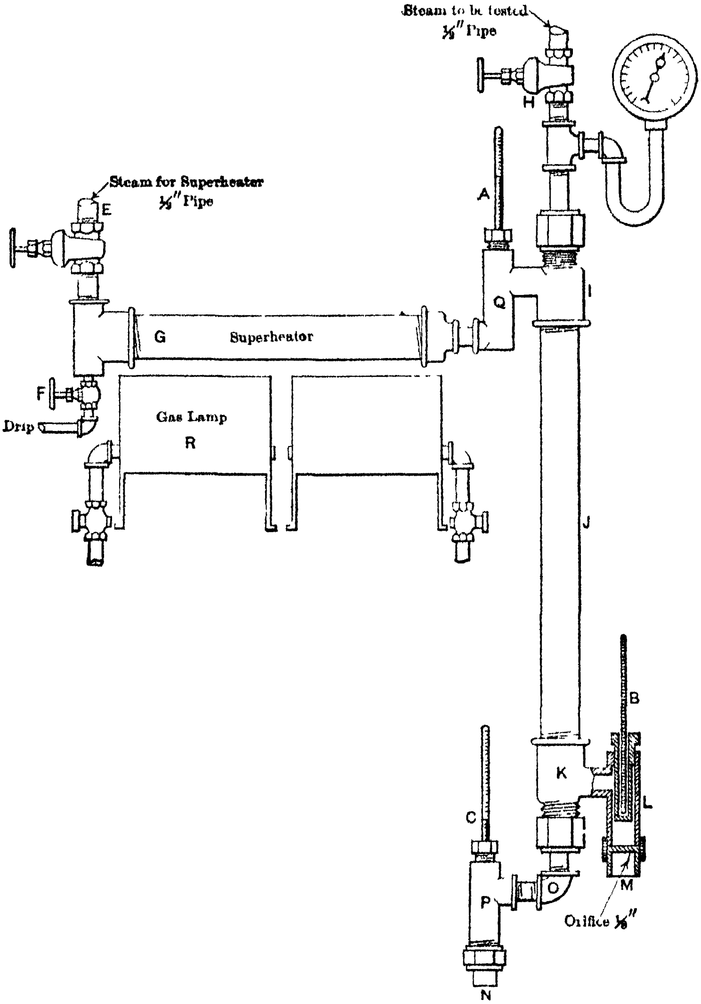

FIG. 383. — BARRUS SUPERHEATING CALORIMETER.

303. The Barrus Superheating Calorimeter is shown in Fig. 383. The pipe carrying the sample steam separates into two branches, of equal size, one, *H*, carrying steam to be tested, the other, *E*, supplying the superheater *G*. *R* is a heater for superheating the steam passing through *G*. From *G* the now superheated steam passes through *Q* and *I* down through the jacket *J*, and out to the atmosphere through the orifice *M*. The test sample passes through the central pipe in *J* and out to the orifice *N*. *M* and *N* are of the same size. Conditions must be such that in passing through *J* the sample steam shall first be dried and then superheated by taking heat from the superheated steam surrounding it, and the latter must still be superheated when it reaches the orifice *M*. Temperatures are taken at *A*, *B*, and *C*, and either the pressure or the temperature of the sample steam must be accurately found.

Theory of the Instrument.—The accuracy of the instrument depends in the first place upon the condition that equal weights of steam must flow out of the orifices *N* and *M* in a given time. If this is not maintained the weights of the two quantities of steam must be considered. This can of course be done by condensing and weighing.

Let W = weight of steam in a stated time from orifice N.
W_1 = " " " " " " " " " M.
T = saturation temperature of steam in main.
T_1 = temperature at A.
T_2 = temperature at B.
T_3 = temperature at C.
R = radiation loss in B.t.u. in the stated time.
x = quality of sample steam.
$1 - x$ = amount of moisture carried by each pound of steam.
C_p = mean specific heat at constant pressure in the range T_1 to T_2.
$C_{p'}$ = mean specific heat at constant pressure in the range T to T_3.

We may then equate the amount of heat lost by the steam escaping through *M* to the amount of heat gained by that flowing out through *N*, or

$$WC_p(T_1 - T_2) - R = W_1[r(1 - x) + C_{p'}(T_3 - T)] \qquad (7)$$

The usual assumption is that $W = W_1$. The mean specific heat $C_{p'}$, for the range T to T_3, can be obtained directly from Fig. 244, while r is taken from the steam-table for the absolute pressure of the sample steam. The factor C_p in the first member of the equation cannot be found directly from Fig. 244; it can, however, be determined in two steps, as follows: Let C_{p_1} be the mean specific heat in the range T to T_1, and C_{p_2} be the mean specific heat in the range T to T_2. Both of these can be read directly from Fig. 244. We may then write

$$WC_p(T_1 - T_2) - R = W[C_{p_1}(T_1 - T) - C_{p_2}(T_2 - T)] - R, \quad (8)$$

and the final form of the equation for this calorimeter is, putting $W = W_1$,

$$C_{p_1}(T_1 - T) - C_{p_2}(T_2 - T) - R = r(1 - x) + C_{p'}(T_3 - T). \quad (9)$$

The radiation loss, R, is some function of T_1, and can be determined by calibration. The following form shows the scheme used at Sibley College for recording the observations made with the superheating calorimeter:

SIBLEY COLLEGE, CORNELL UNIVERSITY — DEPARTMENT OF EXPERIMENTAL ENGINEERING.

LOG OF TESTS WITH SUPERHEATING CALORIMETER.

By...
Date .... .Place.............................
Barometer .... inches Hg

No	Time	Steam Pressure Main.	Absolute Steam Pressure	Temperature.				Moisture in Steam $1 - x$	Quality x
				Saturation Temp of Steam in Main, T.	Jacket Steam Entering T_1	Jacket Steam Leaving, T_2.	Sample Steam Leaving, T_3		

304. The Throttling Calorimeter. — This instrument was designed in 1888 by Professor C. H. Peabody of Boston, and represents a greater advance than any previously made in practical calorimetry.

The equations for its use and limitations of the same were given by Professor Peabody in Vol. IX, "Transactions of the American Society of Mechanical Engineers."

The form of calorimeter at present used in Sibley College was patented by the writer in 1893. Fig. 384 shows the essential details. The sample steam from the main passes through a valve and a throttling orifice into the interior of the calorimeter. Here the low-pressure or exhaust steam completely surrounds a thermometer cup before passing out of the calorimeter through the bottom opening. In many instances a gauge or thermometer cup is inserted between the valve and the throttling orifice in order to determine the pressure of the sample steam as accurately as possible. The pressure of the steam in the calorimeter is obtained by means of a manometer or low-reading gauge, connected as shown. Where the exit from the calorimeter is comparatively free, that is, not obstructed by long pipes, bends, or valves, the so-called back pressure in the calorimeter should not be more than $\frac{3}{4}''$ mercury, and a manometer is therefore the best instrument with which to

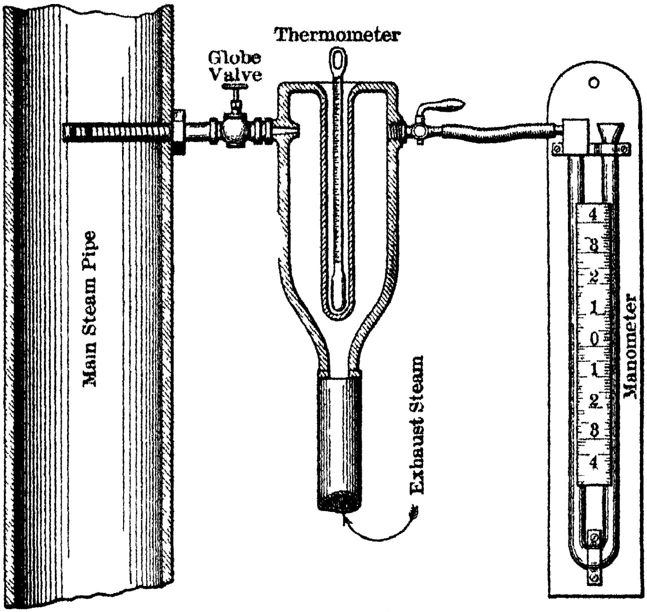

FIG. 384. — THROTTLING CALORIMETER.

determine it. The throttling orifice has a diameter of about .08″. The calorimeter itself is about 3″ in diameter by 6″ long. For the purpose of cutting down radiation the body is nickel-plated, but it is nevertheless advisable thoroughly to cover the instrument with some non-conducting material after installation. The sample pipe leading to the calorimeter should be as short as possible, and should also be thoroughly covered. If the instruments determining the pressure of the sample are inserted between the valve and the throttling orifice, it is essential that the valve be opened wide during a trial.

Theory of the Instrument. — It is assumed in the operation of the throttling calorimeter that there is exactly as much heat in one pound of steam ahead of the throttling orifice as there is in the same weight of steam after throttling; that is, no heat is lost in radiation and no work is done in the expansion process in the orifice. There is, however, less total heat in saturated steam at the lower pressure than there is at the higher, and the excess heat thus liberated in the expansion through the orifice will first evaporate any moisture that the sample contains and will next superheat the dry steam, if any heat is left after the drying process is completed. Hence, provided enough heat has been liberated, the steam in the calorimeter will be superheated. As a matter of fact, this must be the case or the instrument cannot be used, for if the thermometer shows the saturation temperature of the steam in the calorimeter, there is no guarantee that even the original moisture in the sample has been evaporated.

The heat in one pound of the high-pressure steam may be expressed by $(xr + q)$ B.t.u.; that in low-pressure steam by $\lambda + C_p (T_2 - T_1)$ B.t.u., where T_2 is the temperature indicated by the thermometer and T_1 is the saturation temperature of the steam at the absolute pressure existing in the calorimeter. As pointed out, T_2 must be greater than T_1. By the theory above outlined we may then write

$$xr + q = \lambda + C_p (T_2 - T_1), \tag{10}$$

from which

$$x = \frac{\lambda + C_p (T_2 - T_1) - q}{r}. \tag{11}$$

C_p may be closely estimated from Fig. 244.

Example. — Pressure in the main steam pipe is 150 lbs. by gauge, back pressure in the calorimeter 4″ mercury, barometer pressure 29.9″ mercury, temperature in the calorimeter 292° F.

Here we have absolute pressure of sample steam = 150 + 14.7 = 164.7 lbs., absolute pressure in the calorimeter = 29.9 + 4 = 33.9″ Hg. = 16.6 lbs. The saturation temperature, T_1, for the latter pressure, by interpolation from the steam-table, = 218°. The degree of superheat is, therefore, $T_2 - T_1 = 292 - 218 = 74°$. For this range, $C_p = .47$ from Fig. 244. The total heat λ for 16.6 lbs. = 1152.6 B.t.u. Further, for 164.7 lbs., r = 857.9 B.t.u. and q = 338 B.t.u. Finally, by substitution in the above formula,

$$x = \frac{1152.6 + .47\,(292 - 218) - 338}{857.9} = \frac{849.4}{857.9} = 99 \text{ per cent.}$$

In case the result for x is greater than 1.0, it indicates that the sample steam was originally superheated, and the degree of superheat should next be determined as shown in Example 3, page 338.

The following form is now used at Sibley College for recording the throttling calorimeter observations taken and the results:

SIBLEY COLLEGE, CORNELL UNIVERSITY — DEPARTMENT OF EXPERIMENTAL ENGINEERING.

LOG OF TESTS WITH THROTTLING CALORIMETER.

By. ....

Date .... .Place ....

Mean Boiling Temp. in Calorimeter .... Barometer .... . inches Hg.

Number.	Time.	Steam Pressure Main	Absolute Steam Pressure	Steam Temp. Main.	Back Pressure on Calorimeter, inches Hg	Observed Temp. in Calorimeter	Superheat in Calorimeter Degrees	Moisture in Steam	Quality

305. Graphical Solutions for Throttling Calorimeter Computations.

— The Mollier diagram, Fig. 245, offers a very quick means of arriving at the final result for percentage of moisture as based upon throttling calorimeter observations. Thus, in the case of the example above computed, follow the 16.6 lbs. pressure line until it crosses the line of 74° superheat. From this point follow a line parallel to the entropy axis until it cuts the 164.7 lbs. absolute pressure line. This operation assumes a constant heat content of the steam, as laid down in the theory of the calorimeter. The last point of intersection will be found to indicate 99 per cent quality.

This diagram is universal, that is, it takes into account the variation of C_p with pressure and temperature and is therefore applicable to any case.

306. Limits of the Throttling Calorimeter.—When in equation (11) T_2 becomes equal to T_1, there will be no superheat in the calorimeter, and all of the heat liberated in the throttling process has been used to dry the moisture in the original steam. This condition then marks the minimum value of x, that is, the highest percentage of moisture that the calorimeter can indicate under the conditions of initial and final pressure. The following table, which may be computed by substitution in equation (11), assuming $T_2 = T_1$, or may be constructed by means of the entropy heat chart, Fig. 245, shows what the lower limit of the calorimeter is for a series of pressures, taking the pressure in the calorimeter at 15 lbs. absolute. There is, of course, no upper limit, that is, no limit to the degree of superheat in the sample steam that the calorimeter can indicate.

LIMITS OF THE THROTTLING CALORIMETER.

Pressure, pounds per sq in		Maximum per cent of Moisture.	Quality of the Steam, per cent.
Absolute	Gauge (Barometer assumed, 29.9")		
300	285.3	6.7	93.3
250	235.3	6.2	93.8
200	185.3	5.6	94.4
175	160.3	5.3	94.7
150	135.3	4.0	95.1
125	110.3	4.6	95.4
100	85.3	4.0	96.0
75	60.3	3.3	96.7
50	35.3	3.5	97.5

It should be noted, however, that it is not safe to trust the readings from a throttling calorimeter when $T_2 = T_1$, for there is no definite indication under such conditions that the original moisture in the steam has all been evaporated. Further, when T_2 is close to T_1, the readings should be accepted only with caution, and should in fact be accepted only when the thermometer used is known to be correct or when the error is known and allowed for. In the former case some other form of calorimeter must be used, in the latter case the writer would recommend a change if the margin between T_2 and T_1 is less than, say, 10 degrees. In this connection proper lagging of the calorimeter and stem correction for the emerging film of the thermometer are often of importance.

307. Throttling Calorimeter Constructed from Pipe Fittings. — It is possible to construct a very satisfactory throttling calorimeter from pipe fittings, as shown in Fig. 385. Connection is made to the main steam pipe, as explained already elsewhere. The calorimeter is made of ¾-inch fittings arranged as shown; the steam pipe W is of ½-inch pipe, and the throttling orifice is made by forcing a taper plug into the end of the pipe, in which is drilled a hole ⅛ or $\frac{1}{16}$ inch in diameter.

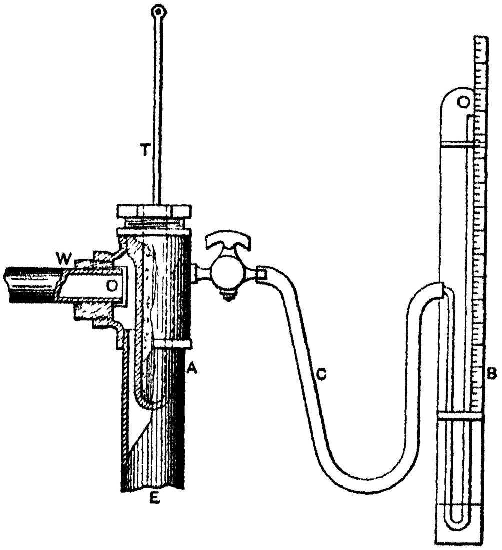

FIG. 385. — THROTTLING CALORIMETER MADE FROM PIPE FITTINGS.

A thermometer cup, Fig. 376, is screwed into the top, and a pet cock inserted opposite the supply of steam. A manometer, *B*, for measuring the pressure is attached by a piece of rubber tubing, as shown. The exhaust steam is discharged at *E*. The back pressure on the calorimeter can be increased any desired amount by a valve on the exhaust pipe; when no valve is used the pressure is so nearly atmospheric that a manometer is seldom required.

308. The Thomas Electric Superheating Calorimeter.— Figure 386 shows a sectional view and Fig. 387 a general view of this instrument.* It consists practically of an electric heater which is constructed by passing a continuous coil of German-silver wire up and down through holes in a soapstone cylinder. Connections to the source of electric supply are made as shown. The steam passes in at *A*, Fig. 386, is dried, and in some cases superheated in passing the coil, and leaves through the opening *C*. The branch pipe *C* contains a section of glass tubing, as shown in Fig. 387. This serves as a

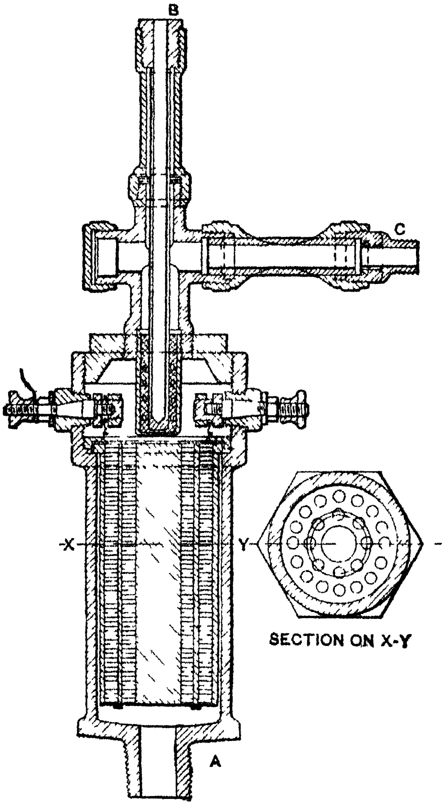

Fig 386. — Section, Thomas Electric Calorimeter

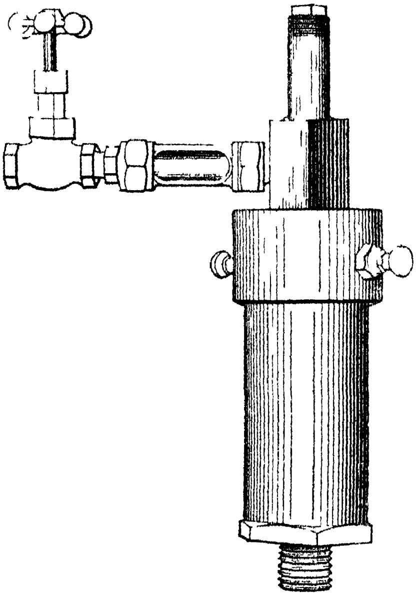

Fig 387 — General View, Thomas Calorimeter

watch glass to indicate whether the steam is wet or superheated. A thermometer is introduced at B, Fig. 386, to indicate the temperature of the steam passing out. Arrangements are made to accurately measure the watts input into the heating coil. The current used is ordinarily 110 volts, about 10 amperes being required.

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* Full description in *Power*, November, 1907.

Operation. — After steady flow is obtained, turn on current and first dry steam, as shown by fog disappearing from watch glass and by thermometer reading. Then superheat to some temperature T. Call input after this temperature has become constant E_T watts.

Next, decrease current until steam is just dry. Call input then E_1 watts. Then $E_T - E_1 = E_2 =$ watts required to superheat T degrees.

It can be shown that the heat required to dry one pound of the steam is then equal to

$$H = K\frac{E_1}{E_2}\text{B.t.u.}, \qquad (12)$$

where K is a constant for any one steam pressure and for a given degree of superheat. This factor has been experimentally determined (see Fig. 388).

Next, if r is the heat of vaporization of the steam at the pressure used, then

$$x = \frac{r - H}{r}. \qquad (13)$$

Example. — Suppose the steam pressure is 165 lbs. absolute, and that to maintain the temperature of the steam at 460° requires 650 watts $= E_T$. When the steam is just dry, as shown by the watch glass, suppose the input $= 260$ watts $= E_1$. Then $E_2 = 650 - 260 = 390$ watts. The critical temperature at 165 lbs. abs. $= 366°$, while $r = 856.8$ B.t.u. Hence, the degree of superheat $= 460 - 366 = 94°$. For this degree of superheat and for 165 lbs. pressure, Fig. 388 shows a value of $K = 59$.

Hence

$$H = 59 \times \frac{260}{390} = 39.4,$$

and the quality is

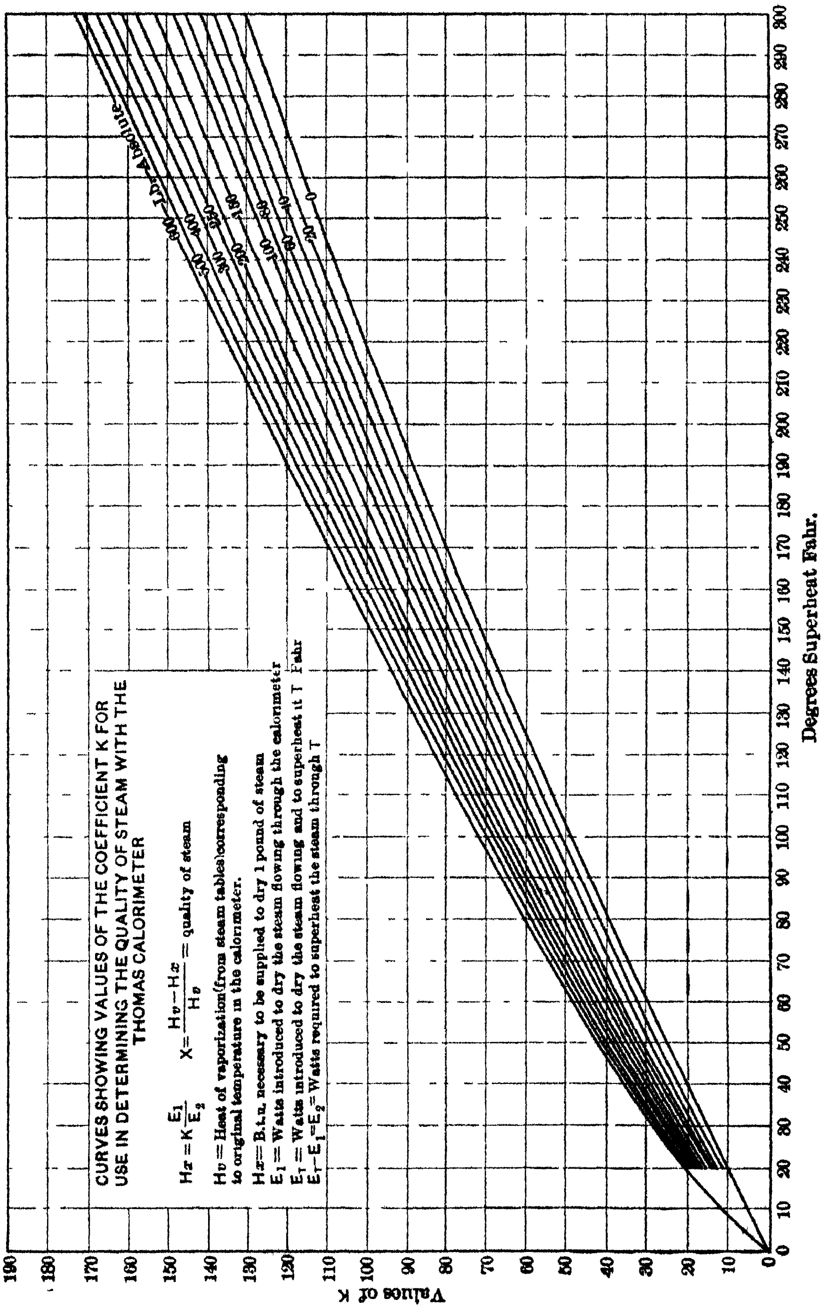
$$x = \frac{856.8 - 39.4}{856.8} = 95.4 \text{ per cent.}$$

309. Methods of Direct Determination of Moisture in Steam. The Separating Calorimeter. — The separating calorimeter is an instrument which removes all water from the sample of steam by some process of mechanical separation, and provides a method of determining the amount of water so removed and also the weight of the sample. This process is dependent upon the greater density of water as compared with that of steam. Thus, for instance, steam at 100 lbs. absolute pressure is more than 260 times lighter than water at the same temperature, and if the sample of steam when moving with considerable velocity can be made to change its direction of motion abruptly, the water will be deposited by the action of inertia.

The accuracy of this instrument depends on the possibility of completely separating the water from the steam by mechanical methods. To determine this, a series of tests were conducted for the author by Messrs. Brill and Meeker with steam of varying degrees of quality. The range in moisture was from 33 to 1 per cent, yet in every case the throttling calorimeter attached to the exhaust gave dry steam within limits of error of observation. The following were the results of this examination:

SEPARATING CALORIMETER.

	Observations on Entering Steam.					Examination of Exhaust Steam from Calorimeter by Throttling Calorimeter		
	T	P	W	w	x	t	x	
Calorimeter	Duration Run, minutes	Gauge Pressure, pounds	Pounds Separated Water in Run	Pounds Condensed Steam in Run	Quality Steam, per cent	Temp in Calorimeter	Quality Steam in Exhaust	No. of Observations
A	25	81.5	1.15	4.45	79.46	281	99.95	6
B	25	78.2	0.15	5.20	97.2	281.3	100.00	6
A	25	80.8	0.525	4.25	89.005	286.5	100.00	6
B	25	79.5	0.150	4.75	96.94	281.8	99.95	6
A	25	78.5	0.300	5.000	94.34	282.8	100.00	6
B	25	77.6	.150	5.45	97.32	282.3	100.00	6
A	24	79.5	1.8	4.55	71.65	280.1	99.94	6
B	24	78.5	1.4	4.90	77.77	279.5	99.9	6
A	20	83.5	1.15	4.1	77.67	286.5	100.00	5
B	20	81.6	1.70	4.75	73.64	282.7	99.98	5
	20	74.8	0.65	3.95	85.87	283.7	100.05	5
	20	82.0	0.85	3.95	82.29	286.8	100.05	5
	20	82.6	0.35	4.15	92.22	285.6	100.0	5
	20	81.5	0.20	3.95	95.15	285.2	100.05	5
A	20	81.4	2.20	4.325	66.28	283.1	100.0	5
B	20	80.3	0.30	4.55	93.81	282.8	100.0	5
A	20	82.0	0.20	4.65	95.8	282.8	99.98	5
B	20	81.1	0.20	4.40	95.7	284.0	100.0	5
Average of 18 trials, involving 98 observations							99.998	



CURVES SHOWING VALUES OF THE COEFFICIENT K FOR USE IN DETERMINING THE QUALITY OF STEAM WITH THE THOMAS CALORIMETER

$Hx = K\frac{E_1}{E_2}$ $X = \frac{Hv - Hx}{Hv}$ = quality of steam

Hv = Heat of vaporization (from steam tables) corresponding to original temperature in the calorimeter.

Hx = B.t.u. necessary to be supplied to dry 1 pound of steam

E_1 = Watts introduced to dry the steam flowing through the calorimeter

E_T = Watts introduced to dry the steam flowing and to superheat it T Fahr

$E_T - E_1 = E_2$ = Watts required to superheat the steam through T

FIG. 188.

This experiment indicates that the complete separation of moisture from steam is possible by mechanical means.

Any radiation in the instrument will increase the apparent moisture in the steam, and must also receive consideration, especially if it be sufficient in amount to sensibly affect the results.

The earliest form of separating calorimeter used in experimental work, in the Sibley College laboratory, consisted of a vessel with an interior nozzle, extending below the outlet and so arranged that the current of steam would abruptly change direction and deposit the moisture into the bottom portion of the vessel. The dry steam was allowed to escape near the top. This instrument, even when covered with hair felt, gave off sufficient amount of heat to sensibly affect the results, and a correction for radiation was essential. The amount of radiation was determined by using two instruments of the same kind and size, arranged so that the discharge from one was the supply to the other.

The second instrument receives perfectly dry steam from the first, the water deposited is due to the radiation loss, which, being the same in both instruments, provides a method of determining its amount. In figuring the percentage of moisture, the amount thrown down by radiation in the second instrument is to be deducted from the total amount caught in the first calorimeter.

In later forms of the instrument the amount of radiating surface has been made so small as to render the correction for radiation, in all ordinary cases, negligible, by constructing the instrument in such a manner as to be jacketed by steam of the same pressure and temperature as in the sample. The form of this instrument is shown in a more or less conventional sketch in Fig. 389, in which the steam is supplied through the pipe *D*, the moisture being received in the interior vessel *E*, the discharge steam passing out of the chamber *E* at the top, into the jacket *F*, and thence out of the instrument through a small opening at *L*, the opening at *L* being made sufficiently small to maintain the pressure in the jacket the same as that in the sample. The discharged steam is then condensed in a can, *J*. This can is provided with a small top in which is set a gauge glass with attached scale, graduated so as to read to pounds and tenths of pounds of water. A gauge glass *N* attached to the calorimeter is provided with index, *mn*, arranged to move over a graduated scale, *S*, which shows the weight of water in the vessel *E* in pounds and hundredths. In using this instrument the condensing can *J* is filled with water to the zero point of the scale. The amount of condensed steam is read on the scale of the can, *J*;

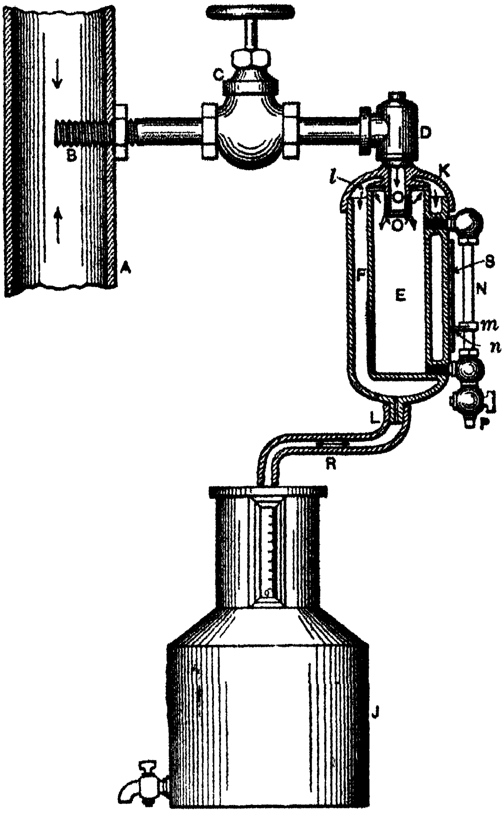


FIG. 389. — SEPARATING CALORIMETER.

the amount of water in the sample of steam for the same time is read on the scale *S*. The percentage of moisture, in case radiation is neglected, is the quotient of the reading of the calorimeter scale *S* divided by the sum of the readings on both scales.

The latest form of the instrument is shown complete with all accessories in Fig. 390, and is a great improvement over the earlier forms in points of portability and convenience. It differs from the form last described principally in the construction of the steam-separating device, which has been increased in efficiency, and in the substitution of a so-called flow gauge attached to the outer jacket, which registers the total flow of steam through the instrument in ten minutes of time.

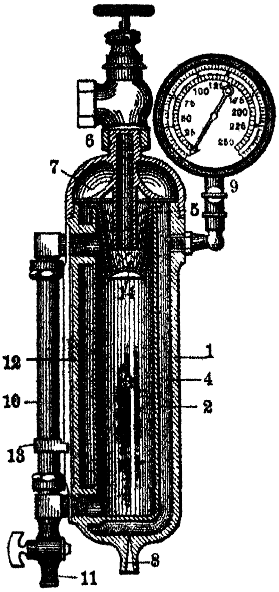


FIG. 390. — SEPARATING CALORIMETER.

The flow of steam through a given orifice is proportional to the absolute steam pressure, by Napier's law* (see page 453), which has been proved correct for pressures above 25 pounds absolute; and hence it is possible to calibrate by trial a pressure gauge in such a manner that the graduations will show the flow of steam in a given time. The only error which is produced in this graduation is that due to changes in barometric pressure, which is never sufficient to sensibly affect the results obtained in the use of the instrument. Much, however, depends also upon the proper construction of the orifice. Unless this is a well-rounded (bell-mouth), smooth opening, the actual discharge of steam will be considerably less than that computed. The quality will therefore appear better than it really is, since w (see next article) will be assumed too large. Should any doubt arise, the accuracy of the readings of the gauge are easily verified by condensing the discharged steam for a given period of time. This should be done occasionally to test the graduations.

310. Formula for Use with the Separating Calorimeter.— Let w equal the weight of dry steam discharged at the exhaust orifice, W the water drawn from the separator, R the water thrown down during the run by radiation. Then the quality of the steam is

$$x = \frac{w + R}{W + w} \qquad (14)$$

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* See "Transactions American Society Mechanical Engineers," Vol. XI., 1887, paper by Prof. C. H. Peabody.

the amount of moisture

$$1 - x = \frac{W - R}{W + w}. \qquad (15)$$

To reduce the radiation loss as much as possible the instrument should be thoroughly covered with hair felt to the thickness of ½ to ¾ inch. In this case the total loss by radiation will be about 0.4 B.t.u.* per square foot per hour for each degree difference of temperature between the steam and the surrounding air. This will amount to about 220 B.t.u. per square foot per hour, or about ⅕ of a pound of steam under usual conditions of pressure and temperature. In the instrument described the actual exposed surface amounts to about $\frac{1}{12}$ sq. ft., so that the condensation loss may be considered as from $\frac{1}{50}$ to $\frac{1}{60}$ of a pound of steam per hour. The total flow of steam through the instrument usually varies from 40 to 60 lbs. of steam per hour, so that if the instrument is covered, the radiation loss would be less than $\frac{1}{20}$ of one per cent. If the instrument be not covered, the loss would be about five times this amount, or under usual conditions about ⅕ of one per cent.

The radiation loss can in every case be determined by using steam of known quality, as determined by the throttling calorimeter, or better still by arranging two separating calorimeters of exactly the same size in series, so that the steam exhausted from the first is used as a supply to the second in a manner already explained.

311. The Limits of the Separating Calorimeter. — The instrument will give correct determinations with any amount of moisture that the sample of steam may contain. With steam containing a very small amount of moisture, the radiation loss will have more effect than with steam containing a greater amount. When the fact is considered, however, that a sample of steam cannot probably be obtained but what differs more than ½ per cent from the average, the futility of making this correction becomes at once apparent. It is, of course, almost superfluous to note that the instrument cannot indicate superheat in the original sample, as the throttling calorimeter is able to do. But the separating calorimeter can be used for degrees of moisture for which the throttling calorimeter fails, and these two instruments together, therefore, cover all of the quality conditions of steam.

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* See numerous experiments, Carpenter's "Heating and Ventilation" (N. Y., J. Wiley & Sons), Chapter IV.

312. General Method of Using the Separating Calorimeter. — The general method of using is given only for the instrument last shown, which is briefly as follows: First, attach the instrument to a pipe leading to the main steam pipe, as already explained, so as to obtain the fairest sample of steam.

Second, wrap the instrument and connections thoroughly with hair felt, to prevent loss of heat by radiation, leaving only the scales visible.

Third, permit the steam to blow through the instrument until it is thoroughly heated, before making any determinations.

Fourth, take the initial and final readings on the scale 12 Fig. 390, at beginning and end of a period of ten minutes of time and note the average position of the hand on the gauge dial during this time. The pressure should be kept as nearly constant as possible during the period of discharge, in which case this hand will remain constant.

Fifth, compute the percentage of moisture as explained by dividing the reading on the scale 12, Fig. 390, by the sum of the readings on scale 12 and the gauge dial.

Attention is again called to the difficulty of obtaining an average sample of steam for the calorimetric determination. The principal cause of this difficulty is due to the great difference in specific gravity of water and steam, as, for example, at a pressure of 100 pounds absolute per square inch a cubic foot of steam weighs 0.23 pound; a cubic foot of water at the corresponding temperature weighs about 56 pounds, or more than 225 times as much. If any great amount of water is contained in the steam, it is likely, if moving in a horizontal pipe, to be concentrated on the bottom; if moving downward in a vertical pipe, to fall under the influence of gravity and inertia; if moving upward in a vertical pipe, it tends to remain at the bottom until absorbed or taken up by the current of steam. The amount of water by weight that will be absorbed as a mist or fog and carried by the steam is not definitely known, but it depends in a large measure on the velocity of flow.

Because of the great difference in weight of water and steam nearly all the water can be deposited from a current of steam, in a vessel or reservoir conveniently connected to the steam pipe, by action of gravity or inertia. Such a device is known commercially as a *steam separator.* The water is removed from the separator either by an automatically controlled pump or trap or by hand.

In the determination of quality it is desirable to remove the free water by a steam separator before making the connection to obtain the sample, as in that case the sample is more likely to be an average one. See papers on this subject in the "Transactions of the American Society of Mechanical Engineers," Vol. XIII, by Prof. D. S. Jacobus, and Vol. XII, by the author.

The following is a convenient form for recording observations and results:

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

PRIMING TEST WITH SEPARATING CALORIMETER.

Made by...191..
Test of..................................Steam..............................
at ....N. Y., ...

No	Time	Duration of Run	Gauge Pressure	Absolute Pressure	Pounds of Water				Moisture.	Quality
					Condensing Can or Flow Gauge		Calorimeter			

313. The Chemical Calorimeter. — This instrument depends on the fact that certain soluble salts will not be absorbed by dry steam, but will be carried over by water, so that if the salt appears in the steam its presence indicates water.

Various salts have been used, but common salt, *chloride of sodium*, gives as good results as any.

The proportion that the salt in a given weight of condensed steam bears to that in a given weight of water drawn from the boiler, is the percentage of moisture in the steam. The method of analysis is a volumetric one, and is as follows:

Add three or four ounces of common salt to the water in the boiler; after it is dissolved, draw from the boiler a small amount of water and condense an equal weight of steam, which are to be kept in separate vessels. Add to each of them a few drops of neutral chromate of potash, but in each case an equal quantity, which amount may be measured by a pipette; the same amount should also be added to a vessel containing an equal weight of distilled water, in order to obtain a standard or zero point for the scale used in the analysis.

By means of a graduated pipette a standardized solution of nitrate of silver is permitted to flow, a single drop at a time, into each of the three solutions. The effect is to cause the formation of the chloride of silver, and until that formation completely takes place the resulting liquid will be whitish or milky; but because of the presence of the chromate, the instant the chloride has all been precipitated the liquid turns red. The amount of nitrate of silver required is measured by the graduated pipette, and gives the information regarding the salt present.

The detailed directions for the test are as follows:

Take in each case 100 cubic centimeters of liquid containing a few drops of neutral chromate of potassium, and use a standard solution holding 10.8 grams of silver to the liter; the following data were obtained in a test:

AMOUNT OF NITRATE OF SILVER REQUIRED TO TURN 100 c.c. RED

100 c.c. of	First Trial.	Second Trial	Third Trial.	
Condensed steam........	0.1 c.c.	0.05 c.c.	0.1 c.c.	*a*
Water from the boiler.....	13.6 c.c.	14.0 c.c.	13 35 c.c.	*b*
Distilled water....	0.05 c.c.	0.05 c.c.	0.05 c.c.	*c*

Let the results with these three samples be denoted by *a*, *b*, and *c* respectively, and the amount of moisture by $1 - x$, we then have

$$1 - x = \frac{a - c}{b - c}. \qquad (16)$$

This gives the following results:

	First Trial.	Second Trial.	Third Trial.
Amount moisture........	$\frac{0.1-0.05}{13.6-.05} = .0037$	$\frac{0.05-0.05}{14.0-0.05} = 0$	$\frac{0.1-0.05}{13.35-0.05} = .00375$

Average = 0.0025.

Instead of common salt, sulphate of soda may be used, and the percentage of moisture determined by the percentage of sulphuric acid present in the steam as compared with that in water from the boiler.

It will be noted that this method of determining moisture is rather complicated, and, on account of their evident limitations, none of the chemical methods are used.

314. Universal or Combination Calorimeters. — It sometimes happens that a throttling calorimeter fails during a test on account of excessive moisture in the steam, and that recourse must be had to some other form, like the separating. If proper arrangements are made it is usually possible to quickly make the change from one to the other, but sometimes this cannot be done. Further, there are cases in which nothing is definitely known about the quality of the steam to be tested, and in such cases the use of a so-called universal calorimeter may simplify matters considerably. These calorimeters are practically the combination of a throttling with a separating calorimeter, these two being used in that order. The best known of these is perhaps the Ellyson, illustrated in Fig. 391.*

No extended description is required. If the throttling at the entrance of the chamber is sufficient to dry and superheat the steam, we have the ordinary throttling calorimeter. If, on the other hand, the steam after passing the opening is still wet, the amount of moisture remaining in it is determined in the lower chamber as in a separating calorimeter. Designating the items connected with the steam in the chamber by the subscript 2, and those for the original sample of steam by the subscript 1, we may write

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* *Power*, Oct. 27, 1908.

$$x_1r_1 + q_1 = x_2r_2 + q_2. \tag{17}$$

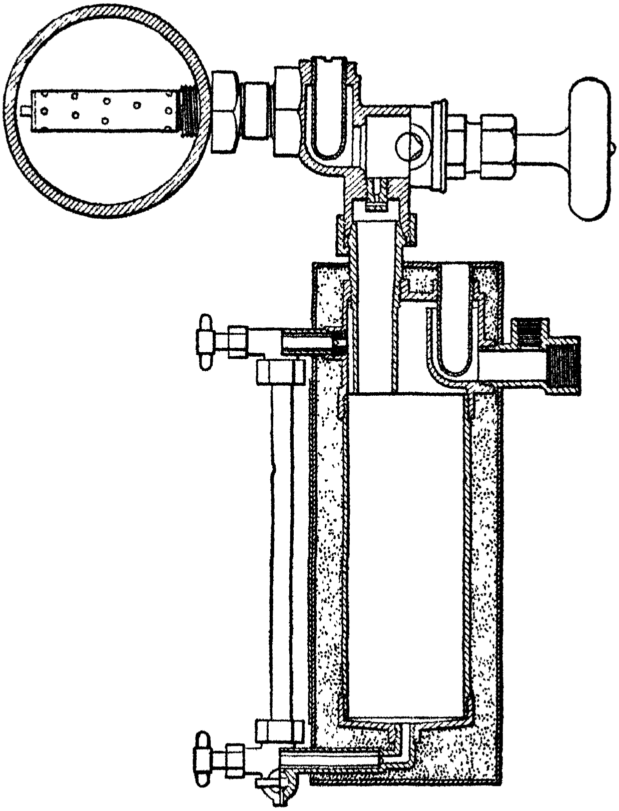

FIG. 391.—ELLYSON UNIVERSAL STEAM CALORIMETER.

In this equation, since x_2 is determined in the separating attachment to the apparatus, all the quantities but x_1 are known, and this may therefore be computed.

315. Comparative Value of Calorimeters. — These instruments, arranged in order of accuracy, are no doubt as follows: throttling; separating; Barrus superheating; Hoadley; continuous condensing; chemical; and lastly the barrel.

The ease with which the throttling and separating instruments can be used, their small bulk, and great accuracy, render them of chief practical importance.

The throttling calorimeter can be used only for steam with comparatively small amounts of moisture, as explained in Article 306; but the separating instrument is not limited by the amount of moisture entrained in the steam. It is not, however, adapted for superheated steam, nor can the results be determined as quickly as with the throttling instrument; when carefully handled the accuracy is, however, substantially the same.

CHAPTER XV.

THE ENGINE INDICATOR.

316. Uses of the Indicator. — The indicator is an instrument for drawing a diagram on paper which shall accurately represent the various changes of pressure on one side of the piston of an engine during both the forward and the return stroke.

The general form of the indicator-diagram for a non-condensing steam engine is shown in Fig. 392; the ordinates of the diagram, measured from the line *GG*, are proportional to the pressure per square inch above the atmosphere; measured from the line *HH*, are proportional to the absolute pressure per square inch acting on the piston. The abscissa corresponding to any ordinate is proportional to the distance moved by the piston. *BCDG* is the line drawn during the forward stroke of the engine, *GEFAB* that drawn during the return stroke. The ordinates to the line *BCDG* represent the pressures acting to move the piston forward; those to the line *GFB* represent the pressures acting to retard or stop the motion of the piston on its back stroke. The ordinates intercepted

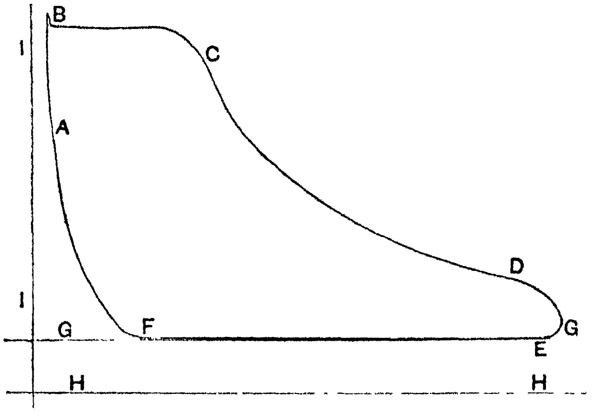

FIG. 392.—DIAGRAM FROM NON-CONDENSING STEAM ENGINE.

between the lines represent the effective pressure acting to urge the piston forward. Since the abscissas of the diagram are proportional to the space passed through by the piston, and the intercepted ordinate to the effective pressure acting on the piston, the area of the diagram must be proportional to the work done by the steam on one side of the piston, acting on a unit of area and during both forward and return stroke.

From an indicator-diagram taken from a steam engine can be obtained, by processes to be explained later: 1. The quantity of power developed in the cylinder, and the quantity lost in various ways, — by wire-drawing, by back pressure, by premature release, by mal-adjustment of valves, leakage, etc.

2. The redistribution of piston pressures at the crank-pin, through the momentum and inertia of the reciprocating parts, and the angular distribution of the tangential component of the piston pressure; in other words, the rotative effect around the path of the crank.

3. Taken in combination with measurements of feed-water or of the exhaust steam, with the amount and temperatures of condensing water, the indicator furnishes opportunities for measuring the heat losses which occur at different points during the stroke.

4. The indicator-diagram also shows the position of the piston at times when the valve motion opens or closes the steam and exhaust ports of the engine. It also furnishes information regarding the general condition of the engine, and the arrangement of the valves, the adequacy of the ports and passages, and of the steam or the exhaust pipes.

317. Indicated and Dynamometric Power. — The engine indicator is used in most engine tests to measure the force of the working medium acting on a unit of area of the piston. A dynamometer of the absorbing or transmission type (see Chap. X) is used to measure the work delivered by the engine. The work is usually expressed in horse power, one horse power being equivalent to 33,000 foot-pounds per minute. The work shown by the indicator-diagram is termed the Indicated Horse Power (I.H.P.); that shown by the dynamometer, Dynamometric or Developed Horse Power (D.H.P.), or Brake Horse Power (B.H.P.).

The mean effective pressure (M.E.P.) per unit of area acting on the piston during one complete cycle is obtained from the indicator-diagram. This quantity, multiplied by the area of the piston in square inches and by the distance traveled by the piston per stroke in feet, will give the work done by the working substance per cycle in foot-pounds. Multiplying by the number of complete working cycles per minute and dividing by 33,000 will give the work done by the working substance in horse power, that is, the I.H.P.

Thus, let p equal the mean effective pressure in pounds per square inch, l the length of stroke of the engine in feet, a the area of the piston face in square inches, and n the number of cycles per cylinder end per minute. Then the work done per minute by the working substance acting on one side of the piston is

$$\text{I.H.P.} = \frac{plan}{33{,}000}. \tag{1}$$

The numerical difference between the I.H.P. and the B.H.P. of an engine must give the power lost in friction, so that

$$\text{I.H.P.} - \text{B.H.P.} = \text{Friction loss}, \tag{2}$$

and

$$\frac{\text{B.H.P.}}{\text{I.H.P.}} = \text{Mechanical Efficiency}. \tag{3}$$

318. Early Forms of the Steam-engine Indicator. — *Watt and McNaught* — The steam-engine indicator was invented by James Watt, and was extensively used by him in perfecting his engine. The indicator of Watt,* as used in 1814, consisted of a small steam-cylinder *AA*, as shown in Fig. 393, in which a piston was moved by the steam-pressure, against the resistance of a spring *FC*. The end of the piston-rod carried a pencil, which was made to press against a sheet of paper *DD*, moved backward and forward in conformity to the motion of the piston. By this method a diagram was produced similar to that shown in Fig. 393.

McNaught's indicator, which succeeded that of Watt and was in general use until about 1860, differed from the form used by Watt principally in the use of a vertical cylinder, instead of the sliding panel, which was turned backward and forward on a vertical axis, in conformity to the motion of the piston.

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* See Thurston's Engine and Boiler Trials, page 130.

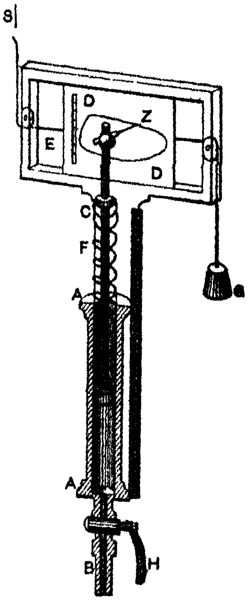

Fig. 393. — Watt's Indicator.

319. The Richards Indicator.* The Richards indicator was invented by Professor C. B. Richards about 1860; it contains every essential constructive feature found in recent indicators, and may be considered the prototype from which all other indicators differ simply in details of workmanship, form, and size of parts.

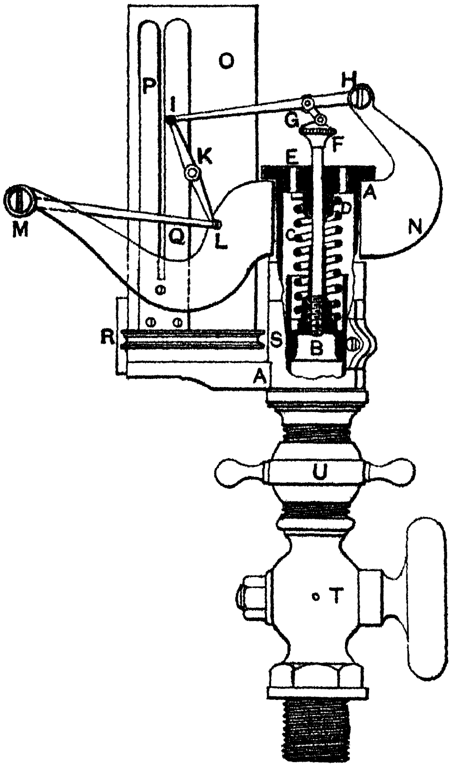

Fig. 394. — Richards Indicator.

The construction of this indicator is well shown in Fig. 394, from which it is seen to consist of a steam-cylinder *AA*, in which is a piston *B*, connected by a rigid rod with the cap *F*. The movement of the piston is resisted by the spring *CD* in such a manner that its motion in either direction is proportional to the pressure. The motion of the piston-rod is transferred to a pencil at *K*, by links which are so arranged that the pencil moves parallel to the piston *B*, but through a considerably greater range. The indicator-spring can be taken out by unscrewing the cap *E*, removing the top of the instrument and unscrewing the piston *B*, and another spring of a different strength can be substituted. The drum *OR* is made of light metal, mounted on a vertical axis, and provided with a spring arranged to resist rotation. The drum is connected to the cross-head or a reducing motion by a cord, and is given a motion in one direction by the tension transmitted through the cord and in a reverse direction by the indicator drum-spring. The paper on which the diagram is to be drawn is wrapped smoothly around the drum *OQ*, being held in place by the clips *PQ*. The indicator is connected to the steam-cylinder by a pipe leading to the clearance-space of the engine, a cock, *T*, being screwed into this pipe, and the indicator connected to the cock by the coupling *U*.

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* See the Richards Indicator, by C. B. Porter; New York, D. Van Nostrand.

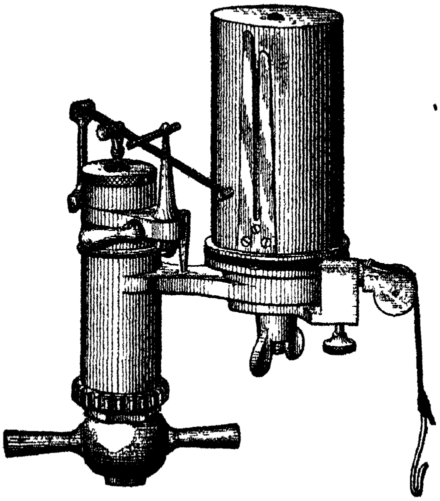

FIG. 395.—THOMPSON INDICATOR.

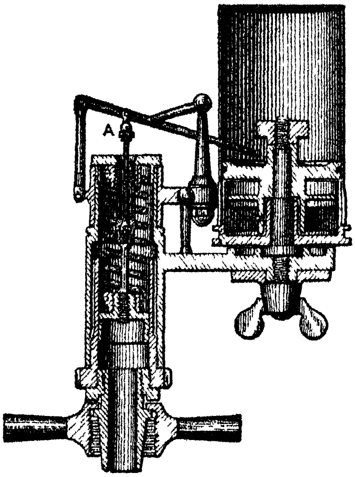

FIG. 396.—SECTION, THOMPSON INDICATOR.

320. The Thompson Indicator.—This indicator is shown in Figs. 395 and 396. It differs from the Richards indicator principally in the form of the parallel motion, form of indicator-spring, and details of workmanship. The parts of the instrument are much lighter, and it is better adapted for use on high-speed engines.

The construction is essentially the same as the Richards; the method of *changing springs* should be thoroughly understood, and is as follows: Unscrew the milled edged cap at the top of the steam-cylinder; then take out piston, with arm and connections; disconnect pencil-lever and piston by unscrewing the small milled-headed screw which connects them; unscrew the spring from the cap and from the piston, substitute the one desired, and put together in same manner, being careful, of course, to screw the spring up firmly against cap and down to the piston-head. The method of changing springs is simple, easy, and convenient, and does not require the use of any wrench or pin of any kind.

The *position* of the *atmospheric line* on the drum may in this indicator be changed by regulating the position of the small screwed head *A*, Fig. 396, on the piston-rod.

321. **The Tabor Indicator.** — The Tabor indicator, shown in Figs. 397 and 398, differs from other indicators principally in producing the parallel motion of the pencil by a pin moving in a peculiarly shaped slot. It also differs in details of construction and in form of the indicator-spring.

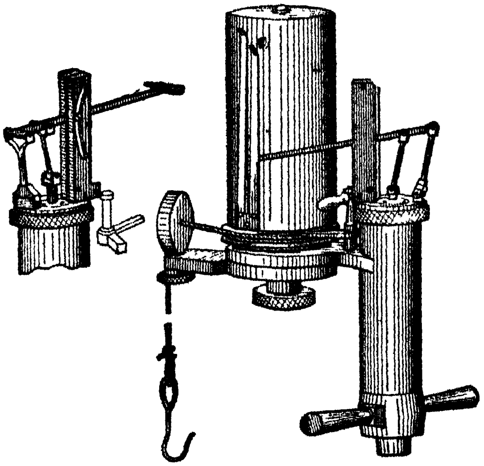

FIG. 397. — TABOR INDICATOR.

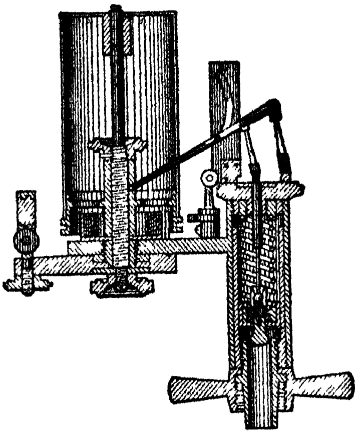

FIG. 398. — SECTION, TABOR INDICATOR

The method of *changing springs* in the Tabor indicator is as follows: Remove the cover of the cylinder, remove the screw beneath the piston, unscrew the piston from the spring and the spring from the cover and replace with the spring desired. When the lower end of the piston-rod is introduced into the square hole in the center of the piston, care must be taken that it sets fairly in the hole before the screw is applied. Unless such care is observed, the corners may catch and cause derangement. The *tension on the drum-spring* may be varied by removing the paper drum, loosening the thumbscrew which encircles the central shaft, lifting the drum-carriage so as to clear the stop, and then winding the carriage in the direction desired.

322. The Crosby Indicator. — The Crosby indicator is shown in Figs. 399 and 400. It differs from those already described in the form of piston and drum-springs and in the arrangement for producing accurate parallel motion.

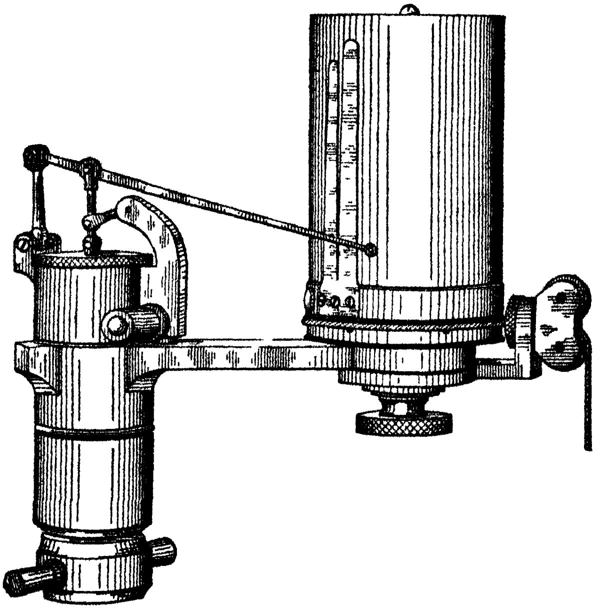

FIG. 399. — CROSBY INDICATOR.

The special directions for this instrument are given by the manufacturers as follows:

To *remove the piston*, spring, etc., unscrew the cap, then, by the sleeve, lift all the connected parts free. This gives full access to the parts to clean and oil them.

To *detach the spring,* unscrew the cap from spring-head, then unscrew piston-rod from swivel-head, then with the hollow slotted wrench unscrew the piston-rod from the piston. To insert a spring, simply reverse this process. Before setting the foot of the spring unscrew *G* slightly, then, after the piston-rod has been firmly screwed down to its shoulder, set *G* up firmly against the bead, and thereby take up all lost motion.

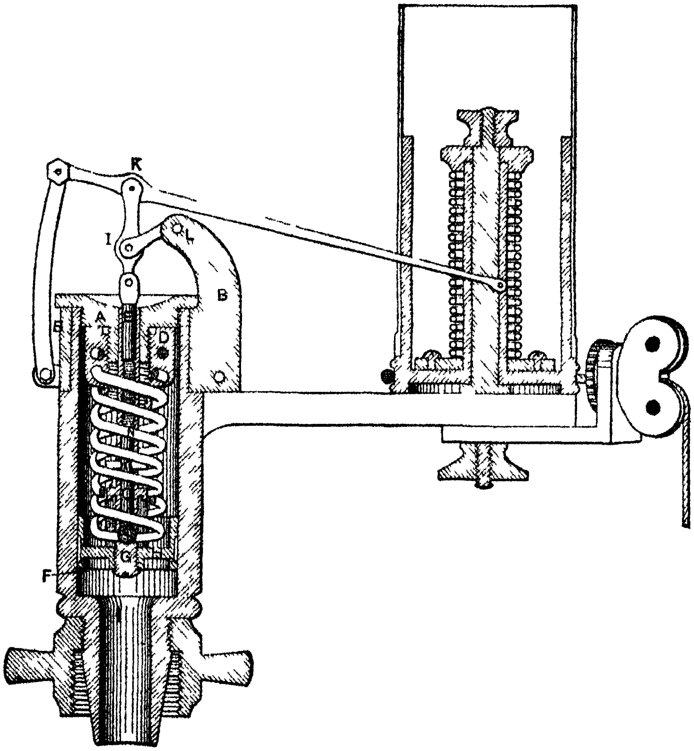

FIG. 400 — SECTION, CROSBY INDICATOR.

It is often desirable to *change the position of the atmospheric line* on the paper. This can easily be done by unscrewing the cap from the cylinder and raising the sleeve *BB* which carries the pencil-movement. Then turn the cap to the right or left, and the piston-rod will be screwed off or on the swivel *E*, and the position of the atmospheric line will be raised or lowered.

Never remove the pins or screws from the joints *K*, *I*, *L*, *M*, but keep them well oiled with refined porpoise-jaw oil, which is furnished with each instrument.

The *tension on the drum-spring* should be increased or diminished according to the speed at which the instrument is used, by means of the thumb-nut on top of the drum-spindle.

323. Indicators with External Springs. — It will be pointed out later more in detail that the varying temperature to which indicator-springs are heated in actual service has a certain effect in changing the scale of spring. This fact has of late years become of greater importance on account of the high temperatures encountered in case highly superheated steam is used and on account of the highly heated products of combustion in gas engines. Provision can be made for cooling the ordinary form of spring, either by using a water-cooled indicator cock or by water-jacketing the spring barrel, but in the past few years indicators with external springs have found their way into the market.

There are two main types of such external spring indicators In the one the spring is flat, is fixed at the end away from the pi barrel of the indicator and is flex over a fulcrum whose position along the spring can be regulated. This type is illustrated by the Bachelder indicator (see Fig. 401). The advantage of this construction, is that the spring is kept cool, and that two or three springs at most can be made to cover the entire range of ordinary pressures, since, by change of fulcrum, one spring may serve for a number of different scales. The main objection to the indicator would seem to be the chances of error involved in setting the fulcrum to the proper point. This of course could be taken care of by calibration. It should be stated that this indicator was on the market some time before the second type about to be described was developed.

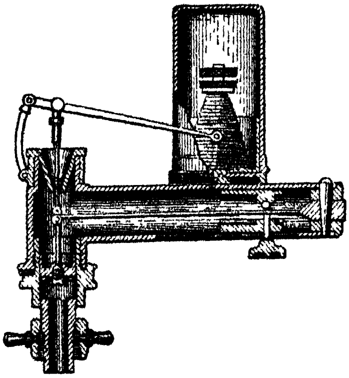

Fig. 401. — Bachelder Indicator.

In the second type the spring is of the ordinary form; in fact, American manufacturers have developed this indicator so that the springs of the ordinary instrument may be used interchangeably.

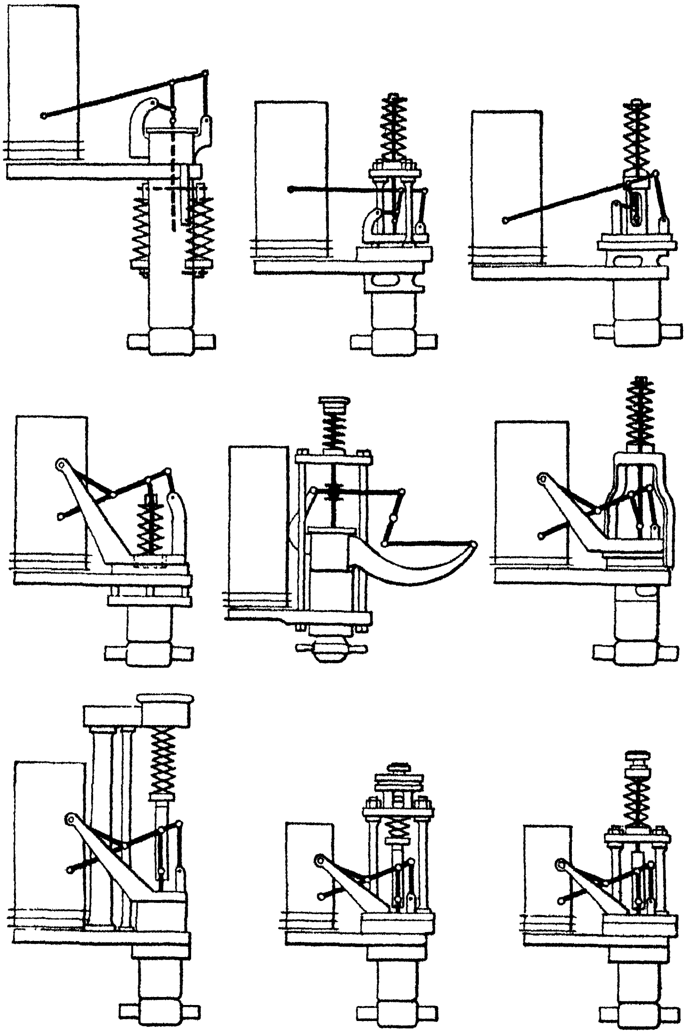

FIG. 402. —VARIOUS TYPES OF EXTERNAL SPRING INDICATORS.

This indicator has for the past six or eight years gone through a course of development during which it assumed a variety of forms, some of which are shown in sketch in Fig. 402.* The problem seemed to be mostly one of reducing the weight of the moving parts without sacrificing strength and necessary rigidity. To-day the springs are used either in tension or compression, and each type of spring has its advocates. It is claimed for the tension spring that it will not buckle over and bind the piston, but it is pointed out also that in the tension spring the force on the piston is transmitted to the outer end of the spring through a rather long and slender rod, which, being under compression, may itself show signs of buckling.

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* Haeder, Der Indicator, p. 13.

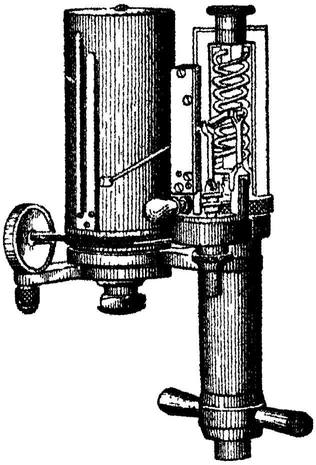

Fig. 403.—Tabor Indicator with External Spring.

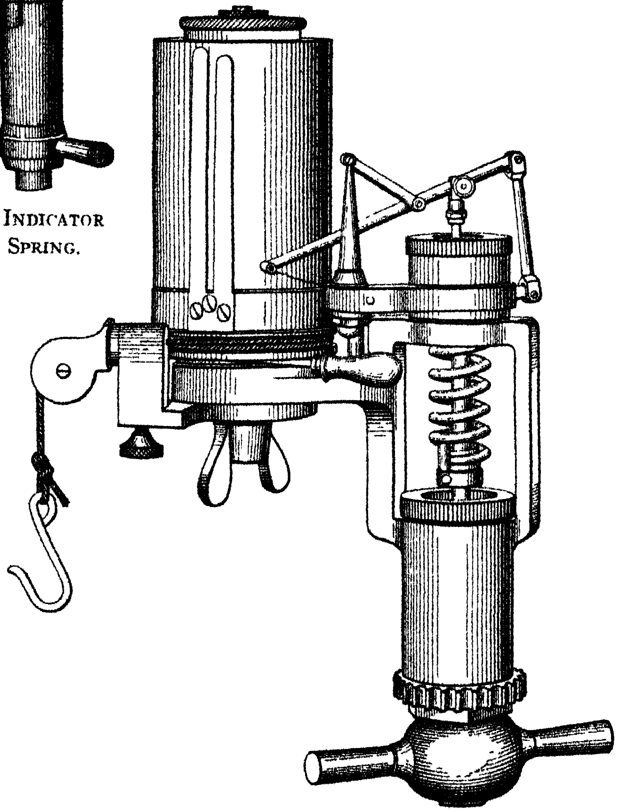

Fig. 404.—American Thompson External Spring Indicator.

Figs. 403 and 404 illustrate two indicators having the compression form of indicator spring. The first one is a Tabor, the second an American Thompson. Fig. 405 shows several views of a Star indicator having the tension form of spring. Another indicator

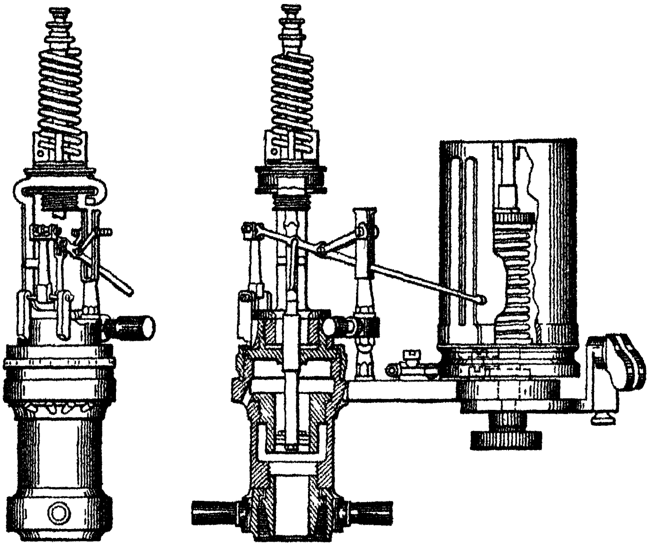

FIG. 405.—STAR INDICATOR WITH EXTERNAL SPRING.

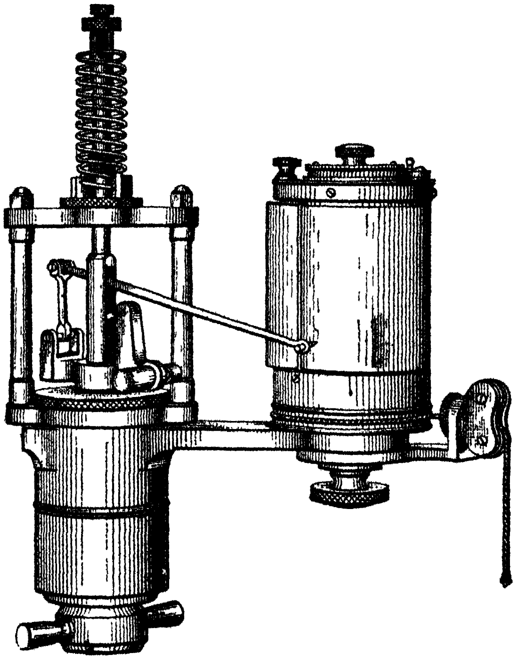

FIG 406.—CROSBY CONTINUOUS INDICATOR WITH EXTERNAL SPRING.

with the same type of spring is the Crosby continuous (see Fig. 406).

324. Indicators for Ammonia Machines. — The indicators manufactured for use with the steam and the gas engine are made of brass or similar material containing copper, and are hence unsuited for use with ammonia or other vapor which attack this metal. For use in testing refrigerating and similar machinery, most of the manufacturers make an all iron or steel indicator, and care should be taken to see that such instruments are used in these tests.

In all the types so far described, the piston must be comparatively loosely fitted in the cylinder to prevent excessive friction and binding at high temperatures, and as a result there is always more or less leakage to the upper side of the piston and from thence into the atmosphere. This is practically nonpreventable in this form, as the upper side of the piston must be freely open to the atmosphere in order that the atmospheric line may be correctly traced and that the piston and spring may operate properly. The leakage of ammonia vapor is naturally not desirable, and to prevent such leakage various forms of indicator have from time to time been produced. The most common among these is the diaphragm indicator, in which a diaphragm takes the place of the ordinary piston spring or piston and spring. This enables complete isolation of the cylinder from the atmosphere, but has the disadvantages that the scale of the spring is not uniform, deflections decreasing as the pressure rises, and that the amount of motion obtainable is so small as to necessitate excessive multiplication in order to obtain a diagram of reasonable height. This process is apt to introduce serious errors if carried out by mechanical linkages in the ordinary way. In one of the optical indicators described below, it is effected in such a way as to introduce only negligible errors.

325. Small Piston Indicators. — The spring which would be required if the ordinary indicator were to be used with very high pressures, such as those attained in gas engines and hydraulic accumulators, might be so bulky that it could not well be fitted into the space available for it. This would mean either the manufacture of special indicators for use with such high pressures, or the making of the ordinary indicator in such a way that springs of usual size could be used with it. Either method means additional expense, in the purchase of two indicators or in the purchase of a number of springs of widely different strengths.

To make the ordinary indicator with the ordinary supply of springs available for use over wide pressure ranges, many of the manufacturers furnish what are known as small piston indicators. In these the indicator cylinder bore is continued downward with a diameter of such size as to accommodate a piston with one-half or one-quarter the area of the ordinary piston. By using the small piston instead of the full-sized one, the ordinary springs can be used for the measurement of much higher pressures. The use of a half-size piston amounts to doubling the scale of the spring; the use of a quarter-size piston is the same as quadrupling the scale.

326. Large and Small Indicators. — In the early days of the indicator the speeds of rotation of engines and the pressures used were both low, but during recent years both have been materially raised. As will be shown below, the errors of indicators increase very rapidly with the speed and with the pressure because of inertia effects, which cause the various parts to over- and under-travel and run ahead or behind their proper positions. These errors can be materially reduced by decreasing the weights and the travel of the moving parts, and most of the makers of indicators now produce a large size for ordinary speeds and pressures and a small size for high speeds and pressures. The small size naturally gives a smaller diagram than the other, but it is found that these small diagrams give results with smaller errors than result from the attempt to get larger ones.

327. The Continuous Indicator. — In some types of engines the successive working cycles traced through a given period of operation vary so radically in amount of energy developed that an average determination of power, obtained by evaluating a series of diagrams taken one at a time and at stated intervals, may be very far from a true mean value. In some cases, as, for instance, in rolling-mill engines, the time of operation is so short that even this procedure is not possible. To take a series of diagrams over one another on the same card is not a satisfactory method on account of the difficulty of integrating all the cards or of locating the mean card. Conditions found in indicating gas engines which govern on the hit-and-miss principle are somewhat similar, although the cycles traced between two miss periods are usually not so numerous or as varying in size as in the case of a rolling mill or hoisting engine. For obtaining the average power developed in such engines with a fair degree of accuracy, the so-called continuous indicator does good service. This is, in principle, just like the ordinary indicator, except that provision is made for moving the paper a certain small distance around the indicator-drum every time the latter oscillates. The means for doing this are various. Sometimes two drums are used, the roll of paper being wound from one onto the other. In most cases, however, two small spools are located inside of the indicator-drum, the paper passing from one spool around the outside of the drum, winding up on the second spool. A modification of this idea is used in the Crosby continuous indicator, Fig. 406.* The small roll seen near the slot in the main drum holds the supply of paper, which after passing around the main drum is wound on a concentric drum inside. The travel of the paper may in this indicator be adjusted so that from 6 to 100 diagrams may be obtained per foot of paper. Fig. 407 shows an example of the work done, the series being taken from a plate-mill engine, starting with the friction load, showing the pass, and ending again with the friction load. The points of admission and of release are marked with the same numbers for each diagram.

328. The Errors of the Piston Indicator. — There has always been a tendency on the part of engineers to regard the indicator as an instrument of great precision, and the fact is that it can be made one of the most precise of any used by the engineer by great care in its selection and operation. As ordinarily handled it gives results which may be in error anywhere from five to as much as ten per cent, and it is doubtful if it ever gives results closer than two per cent, except in the hands of an expert in its use.

As used for the measurement of power, the drum of the indicator should move in synchronism with the piston of the engine, following its changing speed accurately, and the pencil point should move up and down in a vertical line in exact phase with the varying pressure within the cylinder. With modern constructions the pencil motion is theoretically sufficiently accurate to allow one to assume that the pencil point moves vertically and that it multiplies the motion of the indicator-piston by exactly the same amount at every point of its motion. It is never safe, however, in the ordinary case, to arbitrarily assume that the motion of the drum is correct (that is, that the reducing motion is accurate), or that the vertical distances traveled by the pencil point give the pressures within the cylinder correctly (that is, that the scale of the spring is correct).

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* *Power*, Nov. 23, 1909.

The principal sources of error are:

(*a*) Inaccurate springs, that is, springs in which the deflection per unit increase of load is not the same for the entire range of the spring, and springs in which the deflection per unit of load is not what it is assumed to be and what is stamped on the spring. To these errors, which can be more or less closely determined by spring calibration, see Art. 329, must be added that introduced by the uncertainty as to the real action of the spring under the varying temperatures to which it is subjected.

(*b*) Weak springs, that is, springs which are too weak to quickly damp

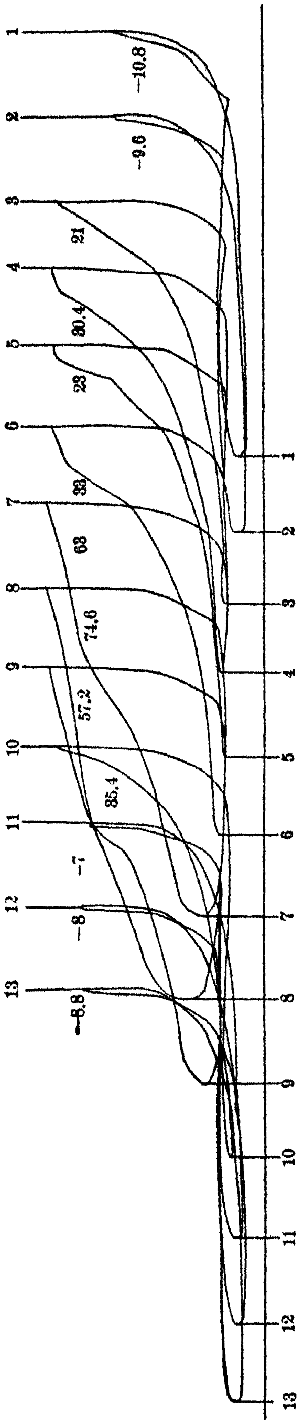

FIG. 407. — DIAGRAMS TAKEN FROM A PLATE MILL ENGINE BY A CONTINUOUS INDICATOR.

out vibrations produced by the inertia effects of the attached parts.

(*c*) Poorly fitted and guided pistons, making the pressure actually applied to the spring something different from that existing in the cylinder below the piston.

(*d*) Improper connections to the cylinder of the engine under test, resulting in the application of pressures below the indicator-piston different from those existing in the engine cylinder.

(*e*) Lost motion in the piston connections and in the joints of the pencil mechanism.

(*f*) Non-parallelism between the indicator-piston travel and the axis of rotation of the paper drum.

(*g*) Inertia effects of indicator-piston, piston connections, and writing mechanism, causing late starting and stopping of vertical motions.

(*h*) Excessive pressure applied to the pencil point when tracing the diagram, resulting in distortion due to the friction of the point on the paper of the drum.

(*i*) Inaccuracies and lost motion in the reducing mechanism used to transmit the travel of the crosshead or other reciprocating part of the engine to the indicator-drum in reduced magnitude.

(*j*) Inertia effects in this reducing mechanism, causing it to lag behind the phase of the engine's motion.

(*k*) Stretch in the string or other material used to connect the reducing mechanism or motion with the drum of the indicator.

(*l*) Inertia effects of the drum and connected parts, causing the moving parts to lag behind the phase of the reduced motion transmitted to it.

All of the sources of error enumerated exist in the application of every piston type of indicator, and it is only by reducing the effect of each to the minimum possible amount that a fair degree of accuracy can be attained. The use of small indicators of light weight and fitted with strong springs is the only satisfactory means of reducing inertia effects, and even these fail at excessively high speeds of rotation, say above five hundred revolutions per minute in the average case. The methods of calibrating for and guard-ing against the other principal errors are given in the following paragraphs.

329. Calibration of Indicator-Springs. — *The Indicator-spring* is usually a helical spring; when in use it has one end screwed to the upper head of the cylinder, and the other screwed to the piston, except in the case of the Crosby spring (see below). To insure accurate results the spring must be accurate, and there must be no play or lost motion between the piston and the cylinder-head, and the spring must receive and deliver the force axially. The number of pounds pressure on the square inch required to move the pencil one inch is stamped on the spring, and the springs are designated by that number, which is called the *scale of the spring*. It is essential to know the error, if any, in the scale. A spring can be readily removed and another substituted when desired; probably the maximum compression should not exceed one-third of an inch. With the usual multiplication of 6 to 1, this would mean a pencil travel equal to $6 \times \frac{1}{3} = 2$ inches, so that the allowable maximum pressure in the case of say an 80-pound spring would be 160 pounds. A rule given by one of the manufacturers is to multiply the scale by $2\frac{1}{2}$ and then to subtract 15 pounds to get the allowable maximum pressure. For the 80-pound spring this would give 185 pounds.

The spring is in many respects the most important part of the indicator, as the form of the diagram is directly affected by any error. The following cuts, Figs 408 to 410, show some of the principal forms adopted by a few of the makers, and it may perhaps be sufficient to state that within the range of action of the indicator any of these forms can be made practically perfect.

The best method of calibrating an indicator spring is still a mooted question. It is admitted by all that the conditions during calibration should as nearly as possible duplicate those during use, but so far no calibrating device which will do this has been produced. Roughly all the methods so far used divide naturally into two classes:

(*a*) Dead-weight calibration, and

(*b*) Fluid-pressure calibration.

Either method may be used with the spring hot or cold.

The term ***dead-weight calibration*** is here used to designate any method in which the pressure is applied to the piston or spring mechanically and the reaction of the spring measured by suitable weighing apparatus.

A convenient and simple apparatus for this purpose is that devised by Professor Cooley of Ann Arbor. In this case the indicator containing the spring to be tested is supported on a bracket above the platform of a platform scale or balance. Force is applied to the indicator-piston by means of a rod which can be lengthened or shortened by turning a hand-wheel. This rod terminates above in a cap nicely fitted to the underside of the indicator piston, while the lower end rests on a pedestal standing on the platform of the balance. Any force applied to compress the spring is registered on the scale beam.

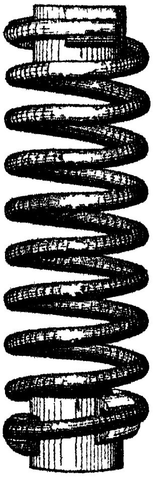

FIG. 408. — THOMPSON SPRING.

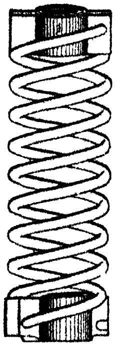

FIG. 409 — TABOR SPRING.

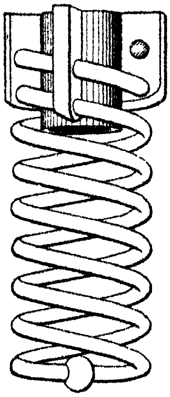

FIG. 410. — CROSBY SPRING.

The reading of the scale beam is that of the force acting on the face of the piston, which is usually some even fraction of a square inch in area, one-half in the ordinary indicator. The spring is made to deflect a certain distance by such a force, but its scale is set in terms of pressure per square inch. Thus, a fifty-pound spring is one which moves the indicator-pencil one inch to indicate a pressure of fifty pounds per square inch, but in an ordinary indicator with a piston having an area of one-half square inch this spring will cause a movement of one inch when an actual load of twenty-five pounds is applied to the face of the piston. It thus follows that the reading of the platform scale would in this case have to be multiplied by two to give a value comparable with the indication of the spring. In general, the reading of the balance must be multiplied by the reciprocal of the area of the indicator-piston in square inches to obtain the load on the piston in pounds per square inch.

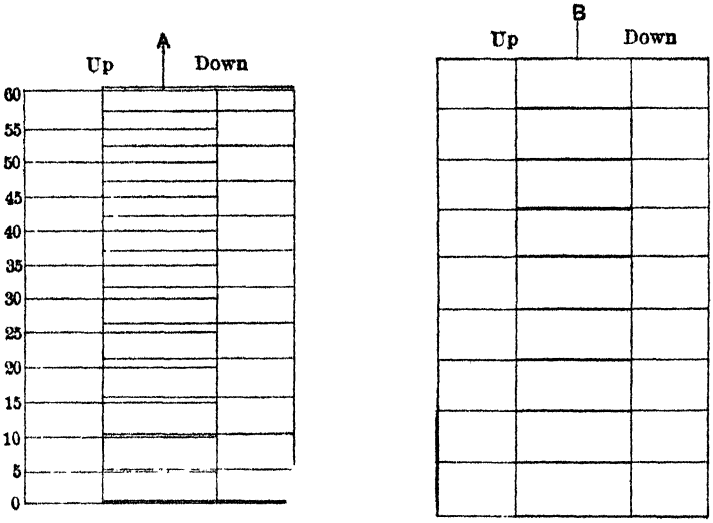

FIG. 411. — SAMPLE INDICATOR-SPRING CALIBRATION DIAGRAMS.

The spring is calibrated by raising the pressure in even increments, generally such as will give pencil positions about a fifth of an inch apart, and rotating the indicator-drum when each desired value is reached, obtaining a line upon the paper on the drum indicating the pencil position for each pressure. After the highest desired pressure is reached the pressure is decreased by similar amounts, marks being made at each value as before. It will generally be found that the two marks obtained for the same pressure value do not correspond (see Fig. 411), those obtained when reducing the pressure being above those obtained when raising the pressure. This is due to friction and similar retarding forces. The average between every two corresponding lines is to be taken as the reading for that pressure. It is very important never to overrun a setting going up or down and then back up to it, as it is evident that the retarding forces will then be acting in the wrong direction.

Another dead-weight tester, a modification of the platform-scale idea, has been lately brought out and patented by Mr. A. B. Calkins.*

Fig. 412 shows that this apparatus consists essentially of a heavy vertical standard, *S*, supporting a compound lever system, L, L_1, L_2. The lower lever L_2 is so arranged that steel scales *F* may be inserted according to the scale of indicator-spring to be tested. Certain weights, *G*, are placed on the poise *E*, also according to the scale of the spring. The weight of the counterpoise *H* is constant. The apparatus is furnished with leveling screws *K* in the base and should be erected on a solid foundation and carefully leveled up. The indicator to be tested is attached to the connection *J′*. The piston *J* fits against the underside of the indicator-piston. The rod of piston *J* is so constructed that its length may be varied by turning the knurled heads *II′*. The lower end of the piston-rod rests against the lever *L*. The method of testing the indicator-spring is as follows: Insert proper scale *F* and place proper weights *G* on the poise *E*, as shown by a card of directions furnished. Place the vernier *C* on the zero of the scale *F*. Attach the indicator to *J′* and adjust the length of the piston-rod so that J is just in contact with the indicator-piston. The proper balance of the apparatus will then be shown by the pointer *B* coinciding with the stationary hair line *A*. In this position draw the atmospheric line. Next run the poise *E* along the scale *F* to any desired position. This will put the apparatus out of balance and throw the pointer *B* to the left of the line *A*. Restore the balance by adjusting by means of *I* and *I′*. Draw a second line on the indicator-drum which will represent the pressure as indicated on the scale *F*. A series of lines at definite pressure intervals going up and down may thus be obtained, as in the apparatus previously described.

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* See *Power* for Aug. 10, 1909.

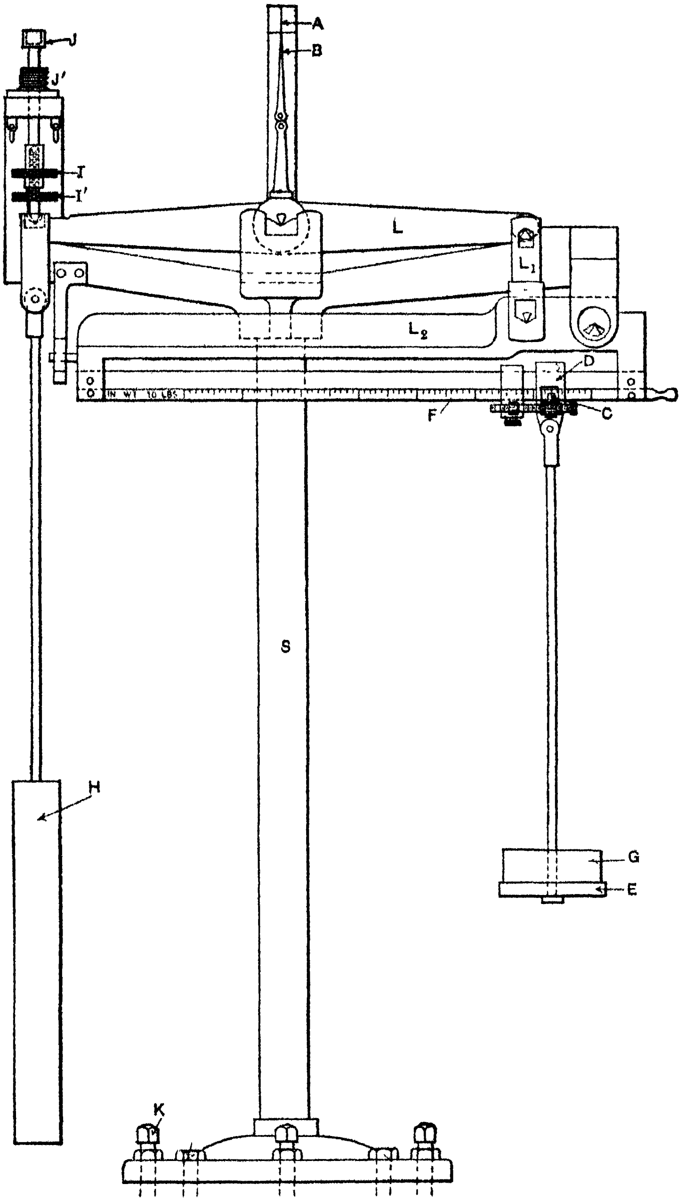

FIG. 412. — CALKINS INDICATOR-SPRING TESTING APPARATUS.

Neither of the above types of dead-weight apparatus is easily adapted for use with steam should it be desired to test springs hot. This can be done in the apparatus shown in Fig. 413, which is of German origin.* In the left-hand figure the indicator is shown loaded by dead weight as if under steam pressure, while the right-hand shows the method of attachment and of loading for vacuum springs. Steam is admitted for heating only, through a lateral connection *A* (see plan view), the method of confining it being shown in the detail figure *B*.

Fluid-pressure calibrating devices nearly always employ steam to load the piston and deflect the spring. Water is not a suitable medium, because it is difficult to maintain any given pressure owing to leakage past the piston. To make the piston sufficiently tight to prevent this would introduce excessive friction and probably make the indicator useless. In general, some dead-weight or equivalent apparatus, as, for instance, a mercury column, is used for measuring the pressure exerted on the piston, because the determination of this pressure by means of a gauge would not give values accurate enough and would, moreover, give values depending upon the accuracy and the calibration of the gauge.

A very accurate arrangement for laboratory use consists of a closed vessel into which steam can be introduced at various controlled pressures and to which can be attached the indicator and a long mercury column. By varying the pressure by convenient increments and determining the value of that pressure from the reading of the mercury column, very consistent and accurate results can be attained. The apparatus has the disadvantage that the mercury column must generally be very long and unwieldy.

A simple device is shown in Fig. 414, consisting of a cylinder, *A*, supported on a bracket above a pair of scales and fitted with a piston having an area of cross-section exactly the same as the indicator-piston. A rod from this piston extends downward onto a platform scale, as shown in the figure. The indicator is connected by suitable piping to the upper end of the cylinder. The steam for the purpose of calibration is adjusted in pressure by a

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* Roser, Prüfung der Indicatorfedern, Zeitschrift d. V. D. Ingenieure, 1902, p. 1575.

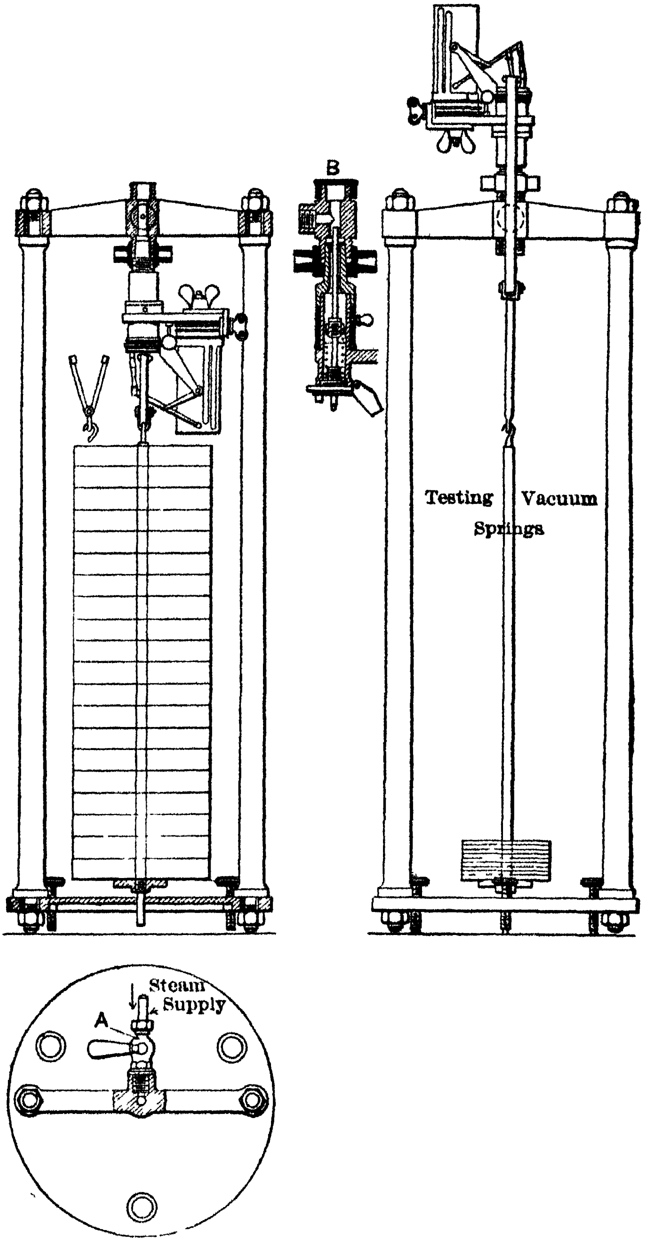

FIG. 413.

valve, *E*, before it enters the drum, *B*. The pressure of the steam in the drum is shown on the attached gauge. This steam pressure exerts an upward pressure on the indicator-piston and a downward pressure on the piston in the cylinder, *A*, which latter, corrected for dead weight, is measured on the weighing-scales shown.

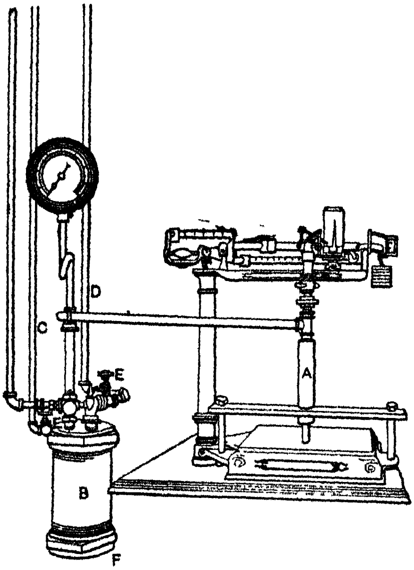

FIG. 414.—INDICATOR-SPRING TESTING APPARATUS.

A modification of this apparatus is shown in Fig. 415, which consists of a vessel, *A*, into which steam can be admitted at any desired pressure. The pressure in the vessel acts on the piston, *K*, which is one-half square inch in area, and it may be measured by the attached scale-beam. The same pressure acts on the indicator-piston. By taking simultaneous readings of the pressure on the piston, *K*, and on the indicator-piston, the calibration may be performed substantially as described. This apparatus has proved satisfactory after an extensive use.

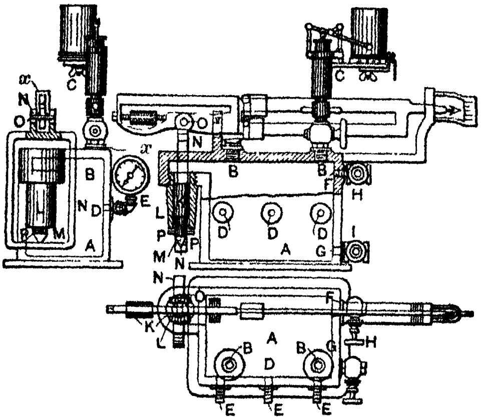

FIG. 415.—INDICATOR-SPRING TESTING APPARATUS.

330. Comparison of Dead-Weight and Fluid-Pressure Methods of Calibration, and Hot vs. Cold Calibration.* The advantages of the dead-weight methods may be outlined as follows:

(*a*) Comparatively low cost of apparatus.

(*b*) Possibility of placing apparatus in any desirable place.

(*c*) Easy and quick manipulation.

Opposed to these are the following disadvantages:

(*a*) The axis of the indicator-piston must be set accurately vertical.

(*b*) The rod transmitting the pressure must be centered accurately on the piston. If either of these items is neglected, eccentric load may introduce excessive piston friction.

(*c*) It is in general not easy to control spring temperature, if the springs are to be tested hot.

(*d*) The scale of the spring depends upon the accurate determination of the diameter of indicator-piston.

Fluid-pressure calibration, unless compressed air is used, generally means the employment of steam, that is, it is practically hot calibration. The great advantage of this method is undoubtedly that the condition of the indicator, pistons and spring approximates more closely to the actual condition of use than if the spring were tested cold. One or two disadvantages of the method may, however, be cited. The first is generally greater cost of apparatus, less quick and easy manipulation, and less choice of location of the apparatus. Further, the temperature which an indicator and its spring attains during use is generally an unknown quantity. It is probably certain that the spring and piston get hotter during calibration than they were during most tests, especially if the apparatus is not quick in its regulation of steam pressure. Tests on spring temperatures have been made, but they do not fully agree, the results showing temperatures anywhere from 20 to 120 degrees less than the temperature of the steam. It should be pointed out here that the fit of the piston has a good deal to do with the temperature of spring and of spring chamber. It has been pointed out, however, that the error introduced by having the spring temperature not quite what it was on the test is generally so small that it comes within the error due to the other imperfections, as, for instance, lost motion, inaccurate straight-line motion of pencil, etc.

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* For detailed information on these points, see the above mentioned article by Roser in the Zeitschrift.

As far as present practice is concerned, American engineers still prefer hot calibration under fluid pressure, i.e., with the use of steam. German engineers, in the adoption of a code, under date of May 5, 1906,* have gone on record in favor of dead weight calibration. The provisions of this code are in brief as follows:

1. Every indicator whose spring is to be calibrated should first be examined with reference to piston friction, fit of piston, and lost motion in the pencil mechanism.
2. Springs are to be tested by dead-weight calibration.
3. Springs should be tested in the indicator to which they belong.
4. Every spring which during use attains temperatures higher than ordinary, should in general be tested both hot and cold, at room temperature and at about 212° F.
5. Springs should be tested at several pressure intervals, at least 5 above the atmosphere and at least 3 below the atmosphere. The report should show the detail results for each interval.
6. The diameter of the indicator piston should be determined at room temperature.

With reference to paragraph 3, the code points out that to test the spring in the indicator to which it belongs is the best way to correct for inaccuracies in the pencil mechanism. While in use, indicators are subject to more or less jar, which serves to reduce the effects of piston friction, and it is consequently advocated to slightly jar the indicator when on the testing stand just before drawing the line for any given pressure interval.

The second calibration advocated under paragraph 4 may be avoided by calibrating cold and then using a temperature correction factor. For accurate work, however, this method can hardly be recommended because the factor is by no means a constant for all springs. The formula that may be used for this computation is

$$p_{t_2} = p_{t_1}[1 - \alpha(t_2 - t_1)]$$

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* See Zeitschrift des Vereins Deutscher Ingenieure under that date.

SIBLEY COLLEGE. — DEPARTMENT OF EXPERIMENTAL ENGINEERING.

Calibration of Indicator-Spring.

Made by

.... *Ithaca, N. Y.*..................................19......

....

....

....

By comparison with ....

Make of Indicator ....

Scale of Spring .... *Spring No.* ....

Actual Pressure on Indicator-Piston		Ordinates				Error
		Inches			Lbs. per sq. in.	
Lbs	Lbs per sq in	Up	Down.	Mean		

By comparison with ....

Make of Indicator ....

Scale of Spring .... *Spring No.* ....

Actual Pressure on Indicator-Piston.		Ordinates				Error.
		Inches.			Lbs per sq in	
Lbs.	Lbs per sq in	Up	Down	Mean.		

where

p_{t_1} = scale of spring at room temperature t_1,

p_{t_2} = scale of spring at some higher temperature t_2,

α = a constant varying from .00019 to .00038, with an average of .00023.

This equation clearly brings out the fact that heating the spring weakens it.

The provision of paragraph 6 is based on the ground that the change of diameter of piston with temperature is not sufficient to appreciably affect the determination of the spring scale.

331. Determination of Scale of Spring and the Evaluation of Diagrams taken with Faulty Springs. — It is always desirable to test springs before they are used on important work. This gives opportunity for rejecting at the outset springs which show serious lack of proportionality. In most cases commercial springs can be had which show but little variation from their nominal scales, and in such cases it is sufficiently accurate to proceed as follows:

From the data obtained (see preceding form for recording) a calibration curve may be constructed with the mean ordinates as determined from test as the ordinates of the curve and the true pressure readings as abscissas. Fig. 416 shows such a curve as constructed from actual test of a 100-pound spring used with a reduced piston (½ size) in the indicator and tested cold. To use this curve, determine the mean ordinate from each indicator-diagram taken on an engine test. The mean effective pressure corresponding to this ordinate may then be read directly from the curve.

The above method results in negligible errors only when the springs are not very far wrong, and above all, when they show satisfactory proportionality, as is the case with the spring for which the data is given in Fig. 416. When the latter is not the case and much depends upon the results of the tests, there are several methods that may be used to correct the results, all of which are, however, more or less laborious.

To illustrate the procedure, take the data contained in the table, page 610, obtained on a calibration test of a nominally 100-pound spring tested hot.

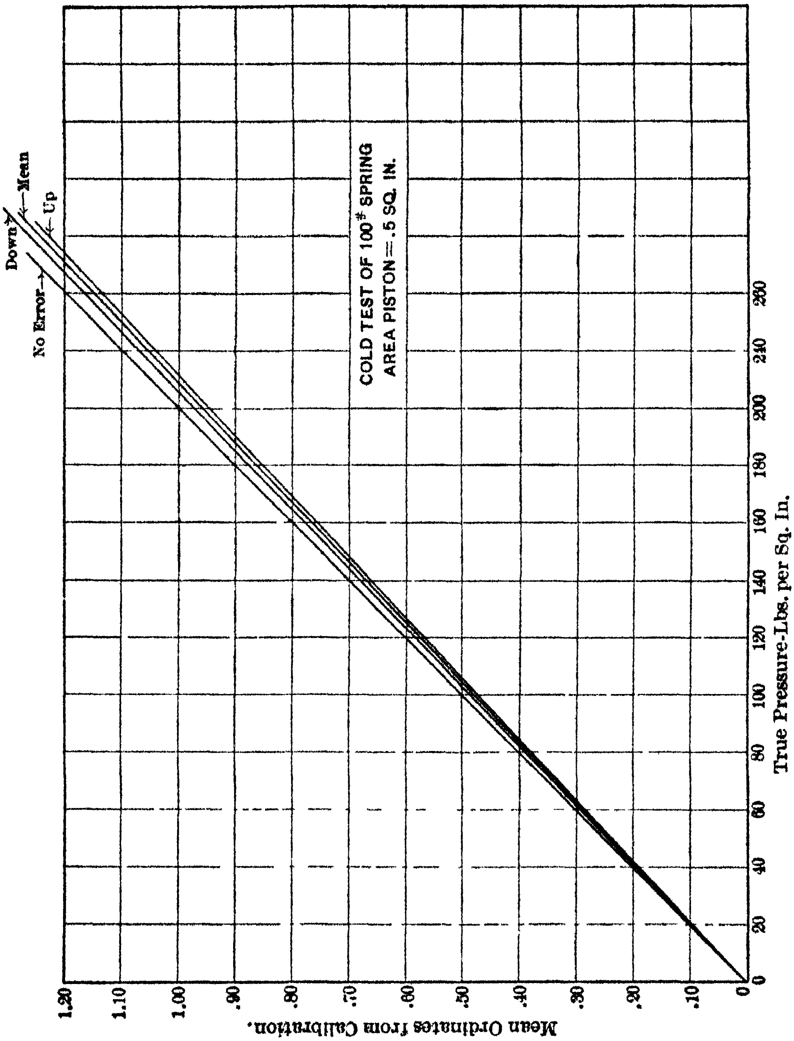

Fig. 416.—Calibration Curve for Indicator-Springs.

This spring shows a decided lack of uniformity, as the scales computed for the individual pressure intervals clearly show. There is not much use in computing the mean spring scale, as is sometimes done, because in this case results varying from 97 to 107 pounds per square inch will be obtained, depending upon the method used. Further, on account of the decided variations in the pencil travels for the same pressure increment, it would be difficult to draw a just mean curve among all of the points.

No. of Reading.	True Pressure in lbs per sq. in. = p	Height of Ordinate Above Atmospheric Line, Mean of Up and Down Readings = h	Scale of Spring for the Successive Intervals. $s = \frac{p}{h}$
1	15	.180	83.4
2	30	.320	93.7
3	45	.465	96.7
4	60	.610	98.2
5	75	.770	97.4
6	90	.910	98.9
7	105	1.050	100.0
8	120	1.175	102.1
9	135	1.315	102.6
10	150	1.445	103.7

In such a case it is best to proceed as follows. Integrate the cards obtained on a test and determine the mean effective pressure for each, assuming the spring correct, i.e., 100 pounds to the inch of pencil travel. Then determine the mean M.E.P. of all the cards and pick out the card nearest the mean. To this card apply the correction following. If the cards do not all show approximately the same size, but if they differ materially, divide them into classes of approximately the same area, and apply the correction method to the mean of each class. This will give in either case a correction factor by which the M.E.P. of each card, as determined on the assumption of a correct spring, can be rectified. The reason for choosing mean cards at all is because it would be too much labor to apply the method to each of a probably large number of cards.

Suppose the diagram, Fig. 417, represents the mean diagram of a set. From the atmospheric line lay off vertically the successive pencil travels which correspond to equal pressure increments, as per table above. Next divide the diagram by lines parallel to the atmospheric line into a number of horizontal strips, as shown.

For each strip determine the mean ordinate in the usual way by planimeter and find the M.E.P. corresponding by multiplying by the spring scale belonging to the particular pressure interval. Thus, for the partial area III, for which the pressure interval is from 105 to 120 pounds, the scale of the spring is, from the table, equal to 102.1 pounds. Finally, the total mean ordinate of the diagram is the sum of the partial mean ordinates of the various strips, and this total mean ordinate divided by the mean ordinate obtained on the assumption that the spring is correct, will give the correction factor to be applied to all of the other diagrams.

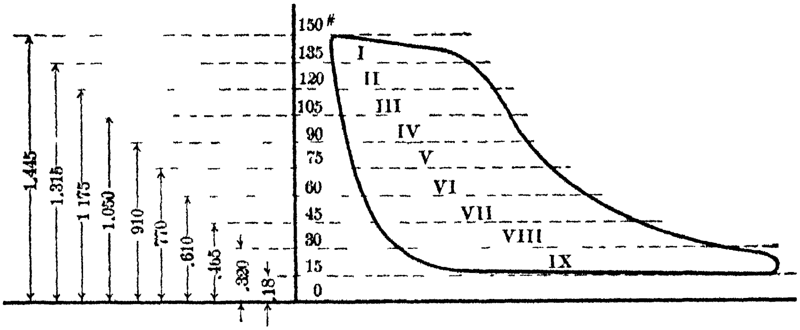

FIG 417 — METHOD OF FINDING CORRECT MEAN EFFECTIVE PRESSURE.

Another method of correction is to completely redraw the mean diagram chosen from the lot and from this determine the proper average M.E.P.

To redraw the diagram, extend atmospheric line (Fig. 418), to left, and along *OA* lay off the successive pencil travels from the table, p. 610. Along *OB* lay off the intervals on the assumption that the spring is a correct 100-pound spring. Determine the line *OC*. Then for any point on the diagram whose ordinate is X_1 we can find, as shown, the corrected ordinate X_2, which determines one point, *D*, on the redrawn diagram. This same operation must be repeated until a sufficient number of points have been found to locate a new diagram accurately. From this the correct mean M.E.P. is then found in the usual way.

It should be pointed out with respect to both of the correction methods above explained, that the work must be very carefully done or else it is worse than useless. It would seem that the last method is open to greater objection on this score than the first.

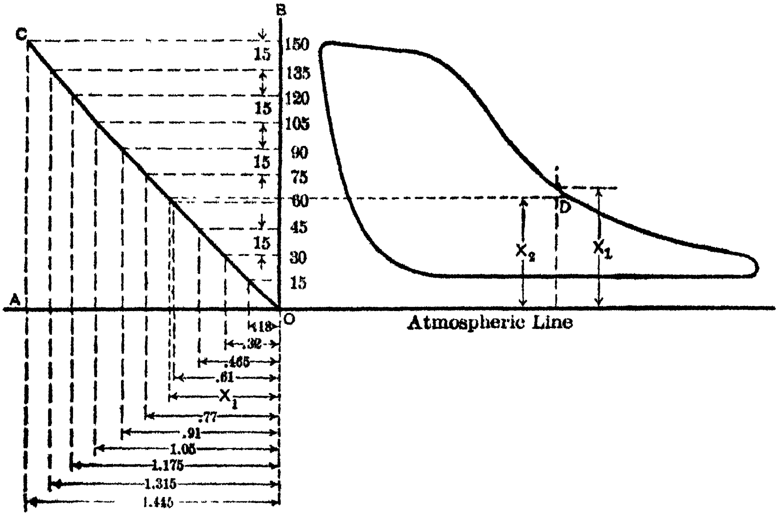

FIG. 418.—METHOD OF FINDING CORRECT MEAN EFFECTIVE PRESSURE.

332. Other Errors in Indicators and Methods of Determining Their Magnitude.—*Inertia Effects Due to Piston and Pencil Motions.*—In theory, the pressure exerted by the spring should at all instants balance the pressure exerted by the working fluid on the piston. In practice, this may or may not be the case, because of the inertia of the indicator-piston and of the pencil mechanism.

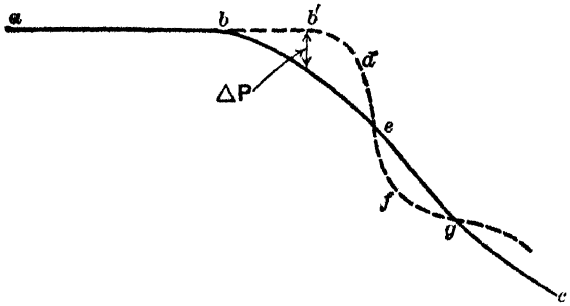

FIG. 419.

Suppose that the pressure of the working medium has been constant for some little time and that it then steadily decreases according to the law of the line *abc*, Fig. 419.

The pencil has followed the line *ab*, fluid pressure and spring force balancing. Now, however, the pencil refuses to follow *bc*, because a certain drop in pressure is necessary to overcome the piston inertia, even if we assume that the indicator is without friction. Hence the pencil draws line *b b′* until the pressure difference is some value ΔP. The piston now commences to move downward, but, although at *d* the piston may drop as fast as the pressure, at that point the pressure difference is greater than ΔP, and hence the piston receives an *acceleration* downward. At *e* the forces apparently again balance, but owing to its momentum the indicator-piston then overshoots the mark, repeating the same thing below the actual pressure line along *efg*.

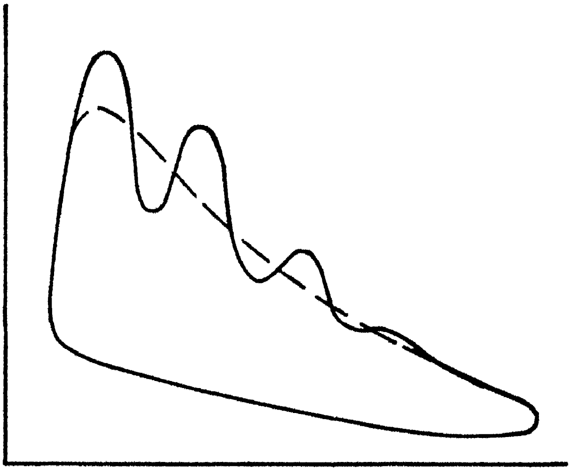

FIG. 420. — DIAGRAM SHOWING INERTIA EFFECTS.

Friction of the piston, not excessive but ordinary, modifies this action in that it acts as a dampener, and in most cases soon reduces the vibrations to zero.

This action may often be observed in indicating a gas engine, because the range of pressure is high and the pressures rise and fall with great rapidity (see Fig. 420). A change to a stiffer spring is sometimes of benefit in toning down the amplitude of these vibrations, but unless these are excessive, their occurrence should rather be taken as a sign of a free working instrument.

Effect of Excessive Friction or Sticking of Piston, or Striking of the Pencil Mechanism. —The effects produced by some of these faults are in some cases very similar to inertia effects, and the two may sometimes be confused. Thus, the waves in the expansion line of the diagram in Fig. 421 were most likely due to sticking of the indicator-piston rather than due to inertia, especially since there is no such effect noticeable at the beginning of admission at *a*. To help distinguish between the two effects, it may be said

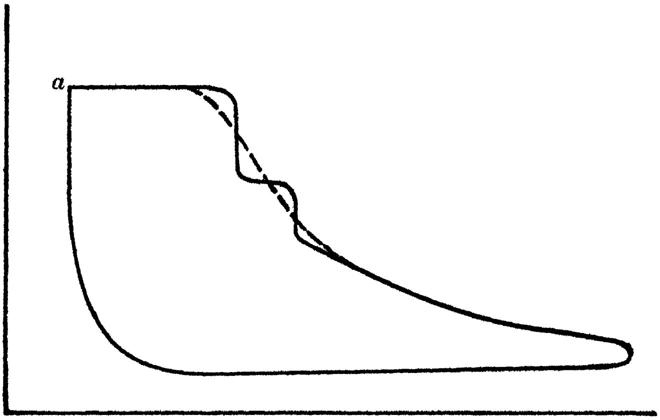

FIG. 421.—DIAGRAM SHOWING EXCESSIVE PISTON FRICTION.

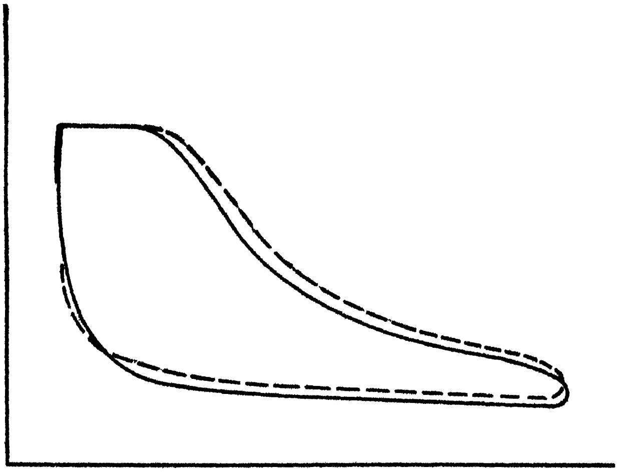

FIG. 422.—EFFECT OF EXCESSIVE PENCIL FRICTION.
NORMAL DIAGRAM IS FULL LINE.

that excessive friction or sticking of an indicator-piston sometimes affects the compression as well as the expansion line, while inertia effects only rarely show in the compression line. Further, inertia effects should be nearly harmonic vibrations, and if the wavy lines are redrawn to abscissas representing equal intervals of time, the curve obtained should be of the nature of a sine curve.*

Effect of Excessive Pencil Friction. — Pressing the pencil too strongly against the paper gives a faulty diagram, the main effect being to make all events late, (see Fig. 422).

Effect of Drum Striking the Stops. — This occurs usually at the inner stop, when the string is too long; too short a string may simply pull in two. The effect is to cut a piece off the end of the card, as shown by the shaded areas in Figs. 423 and 424. In some cases this effect may be hardly noticeable and may escape observation. It is best in every case, after an indicator has been applied and set in motion, to put one finger lightly on the top of the drum, when any striking of the drum will at once manifest itself by a distinct jar.

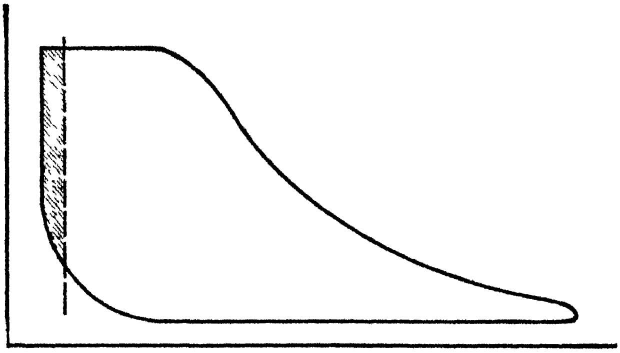

FIG. 423. — EFFECT OF DRUM STRIKING THE STOPS.

Test for Parallelism of the Pencil-movement with respect to the Axis of the Drum. — Parallelism between movement of pencil and axis of drum may be tested for by removing the spring from the indicator, rotating the drum, and drawing a horizontal line; then holding the drum stationary in various positions press the piston of the indicator upward throughout its full stroke, while the pencil is in contact with the paper. The lines thus drawn should be parallel to each other and perpendicular to the horizontal line.

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* See article by Thomas Hall, *Power*, August, 1906.

Tests of Drum-Motion and Adjustment of Drum-Spring. — The accuracy of the drum-motion depends on the form of the drum-spring, the mass moved, the length of the diagram, and the action of the connecting cord.

Indicator-drums would revolve in a harmonic motion if the inertia of the mass could be neglected. The speed of rotation is greatest near the half-stroke of the piston; therefore, if the drum-spring tension can be adjusted so as to exactly counterbalance the effect of the inertia of the moving parts, the theoretical harmonic motion will be nearly realized.

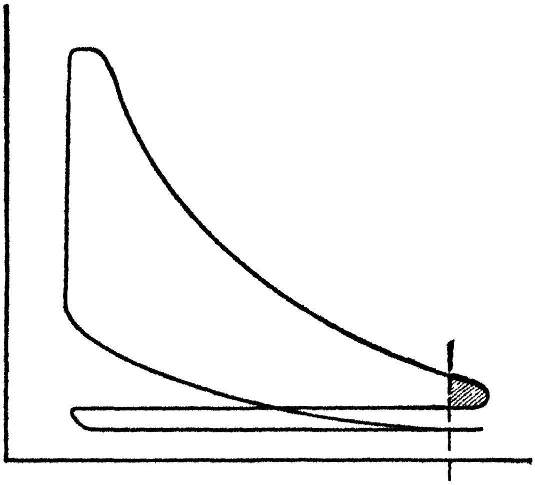


FIG. 424. — EFFECT OF DRUM STRIKING THE STOPS.

In most indicator drum-springs the tension increases directly in proportion to the extension. Since the speed of the drum is greatest at half-stroke, at this point the drum will run ahead of its theoretic motion if the spring tension is not sufficient to counteract the effect of the inertia of the moving parts. Therefore if the tension of the drum-spring is adjusted to exactly balance the effect of inertia at half-stroke, the card should be as nearly as possible theoretically correct. To obtain the value of this tension is not a simple matter. It depends upon the speed of rotation, the mass of the rotary parts, the initial drum-spring tension, the weight and stretch of the connecting cord, and the friction of the mechanism. The initial spring tension must of course be such that on the return stroke the spring can accelerate the drum at least as fast as the engine piston returns. Otherwise the string will at some part of the travel become slack, which leads to non-permissible errors.

Concerning the stretch of the cord, care should of course be taken either to use connections as short as possible, or to use a material, like very fine piano wire, which will not stretch. The objection to such materials in general is their excessive weight. If the length of the cord, or its quality with regard to stretch, is such that the error is not serious, and if the tension on the cord does not vary a great deal, it can further be shown that the action of the deformation existing is such as not to disturb the proportionality between the motion of the piston and that of the drum, except for the action of friction that may exist. Hence in many cases the effect of the error introduced by stretch of cord becomes negligible.

It must be evident that a mathematical determination of proper drum-spring tension becomes practically useless on account of factors in the problem which are peculiar to each individual case. It is better, therefore, where it becomes necessary to know something about the error introduced in the diagram by inertia of moving parts and by varying tension in the cord, to rely upon actual tests.

The total error introduced by inertia can be determined as follows: Attaching the indicator to an engine, permit it to run sufficiently long to harden the cord and the knots, then stop the engine, turn it over by hand, and find the length of the diagram with the speed so small as to eliminate the inertia; leaving the cords connected, run the engine at full speed: any inertia effect will be shown by an increase in the length of the diagram. This increase in length may be due partly to stretch in the indicator-cord caused by inertia of the rotating parts, and even with the best tension on the springs it may be sensibly lessened by the use of wire. A simple arrangement, consisting of a pin and connecting-rod leading to the faceplate of a lathe, the tool-rest being utilized as a guide, may be used instead of an engine for obtaining complete determination of this error. The amount of error caused by over-travel of the drum has been found by experiment to be from 0.5 to 1.5 per cent at 250 revolutions, with the best tension on the drum-spring.

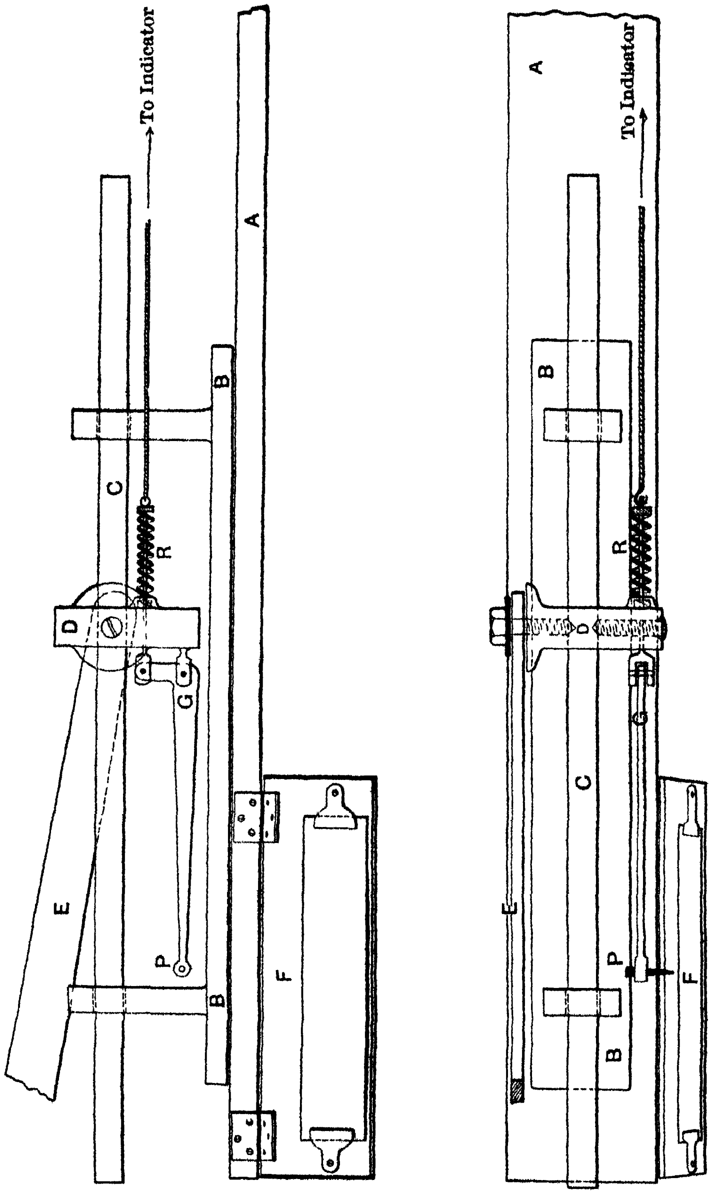
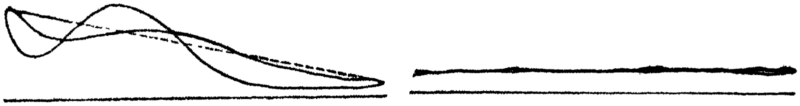


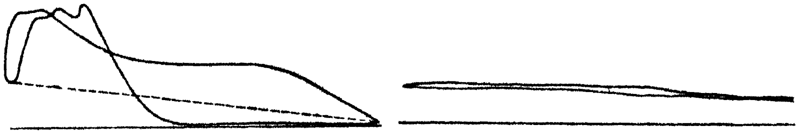
FIG. 425.—APPARATUS FOR TESTING DRUM-SPRING TENSION.

It is often important to determine whether the drum-spring maintains a uniform tension on the cord, or whether it alternately exerts a greater or less stress. This may be determined by the instrument shown in Fig. 425. The apparatus consists of a wooden plate, *A*, on one end of which is fastened the brass frame, *BB*, carrying the slide, *C*, with its crosshead, *D*. The head of the spring, *R*, is screwed to the crosshead, while the other end is connected with the bent lever, *G*, carrying a pencil *P*. The connecting-rod, *E*, which moves the slide, *C*, receives its motion from a crank not shown in the figure. The swinging leaf *F* holds the paper on which the diagram is to be taken. The indicator to be tested is clamped to one end of the plate *A*, and the drum-cord connected with the free end of the spring. The crank is made to move at the speed at which it is desired to test the drum-spring. The paper is then pressed up to the pencil and the diagram taken. If the tension on the cord is constant, the lines which represent the forward and return strokes will be parallel to the motion of the slide; but if the stress is not constant, the pencil will rise and fall as the stress is greater or less. The line drawn when the cord has been detached from the indicator (Fig. 426) is the line of no stress. In the diagram, horizontal distance represents the position of the drum, and vertical distance represents strain on the cord. The perfect diagram would be two lines near together and parallel to the line of no stress, and would represent a constant stress, and consequently a constant stretch of the cord.



Indicator. 250 Revolutions
A

Indicator. 250 Revolutions
B



Indicator. 400 Revolutions
C

Indicator. 400 Revolutions
D

FIG. 426. —DIAGRAMS SHOWING VARIATION IN DRUM-SPRING TENSION.

When the length of the cord and the amount it will stretch under varying stresses are known, the errors in the diagram due to stretch of cord caused by irregular stresses applied by the drum-spring can be calculated.

333. Method of Attaching the Indicator to the Cylinder. — Holes for the indicator are drilled in the clearance-spaces at the ends of the cylinders in such a position that they are not even partially choked by any motion of the piston. These holes are fitted for connection to half-inch pipe: they are located in horizontal cylinders preferably at the top of the cylinder; but if the clearance-spaces are not sufficiently great they may be drilled in the heads of the cylinder, and connections to the indicators made by elbows. The holes for the indicator-cocks are usually drilled by the makers of the engine, but in case they have to be drilled great care must be exercised that no drill-chips get into the cylinder. This may be entirely prevented by blocking the piston and admitting twenty to thirty pounds of steam pressure to the cylinder.

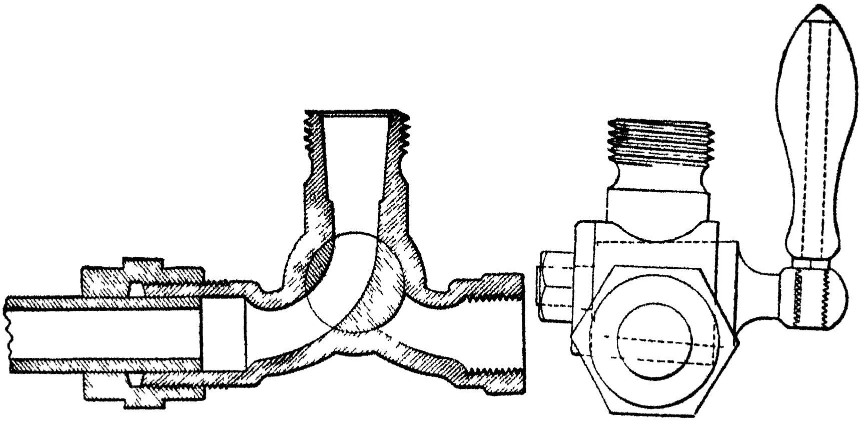
The connections for the indicator are to be made as short and direct as possible. Usually the indicator-cock can be screwed directly into the holes in the cylinder, and an indicator attached at each end. In case a single indicator is used to take diagrams from both ends of the cylinder, half-inch piping with as easy bends as possible is carried to a three-way cock, to which the indicator is attached. The cock is located as nearly as possible equidistant from the two ends of the cylinder.

The form of the three-way cock is shown in Figs. 427 and 428 and the method of connecting in Fig. 442.

In connecting piping for indicators, use lead or oil and graphite on the threads as sparingly as possible, as an excess of these materials is likely to get into the indicator and prevent the free motion of the piston.

For accurate work an indicator should be used at each end of a double-acting cylinder rather than the pipe connections and three-way cock described. Further, all connections must be of practically uniform bore, as short as possible, and free from sharp bends. Indicators seem to work just as well in a horizontal as in a vertical position, so that it is sometimes advisable to place the indicator horizontally rather than to use an elbow or bend. Care must be taken to see that the connections used are sufficiently rigid to prevent oscillatory motions of the instrument under the action of the rapidly changing forces brought into play. Such motion would of course vary the relative position of the indicator and engine crosshead and would correspond to the use of a string of varying length.

Errors of such magnitude as to cause an error of five per cent in the area of the diagram are easily introduced by improper connections, and tests have shown that by sharp bends, long pipes, and restrictions of bore, errors of as much as twenty per cent can be introduced.



FIGS. 427 & 428. — THREE-WAY COCK FOR INDICATOR CONNECTIONS.

334. Reducing-motions for Indicators. — The maximum motion of the indicator-drum is usually less than four inches; consequently it can seldom be connected directly to the crosshead or other reciprocating part of the engine, but must be connected to some apparatus which has a motion less in amplitude but corresponding exactly in all its phases to that of the engine piston. This apparatus is termed a *reducing-motion.* Since the horizontal components of the indicator-diagram, and consequently its area and form, depend upon the motion of the piston, it is evident that the accuracy of the diagram depends upon the accuracy of the reducing-motion. Various combinations of levers and pulleys have been used for reducing-motions, a few of which will be described. Several simple forms of reducing-motion are given here as suggestions, but it is expected that the student will devise other motions if required, and ascertain the amount of error, if any, in the motion used.

There are two main types of reducing-motions to be considered, —*pendulums* (including pantographs) and *reducing wheels.*

In the case of *pendulum reducing-motions,* it is of special importance to see that the angle made by the cord as it passes over the indicator guide pulley at one end and as it leaves its point of fastening at the reducing-motion at the other end, does *not* change its magnitude during the entire range of motion. Just what this means is perhaps best illustrated by a diagram, Fig. 429.

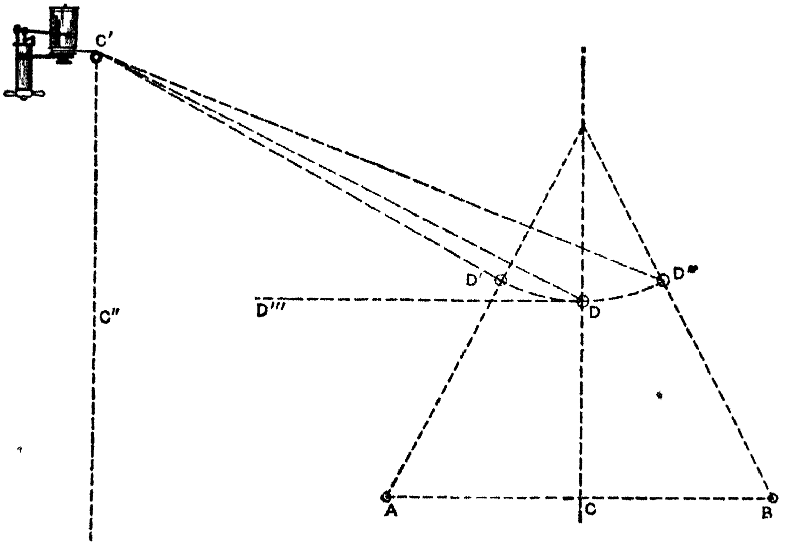


Fig. 429.

Here A–B represents the line of travel of any reciprocating part of the engine which travels in unison with the piston. To lead a cord directly from a pin D on the pendulum to the pulley C' on the indicator would give an incorrect motion because the angle changes from $D\ C'\ C''$ at midstroke to $D'\ C'\ C''$ at one end and to $D''\ C'\ C''$ at the other end of the stroke. It must be obvious that the paper travels a much greater distance while the piston moves from A to C

than when it moves from *C* to *B*. The conditions are purposely exaggerated to make this clear. To minimize the error in this construction the string should lead off in the direction *D* D''' and a guide pulley should be placed somewhere along this line as far from *D* as possible.

The correct remedy for the trouble outlined in Fig. 429 is to use a sector grooved on the edge, which must be fastened to the lever so as to have its center of rotation at the lever pivot, and the arc must be of such length that at all times the cord will be tangential to the sector. Fig. 430 shows conventional sketches of pendulum motions with the sector in three different positions. Sometimes, instead of the sector a circular disk is used, which may either be turned to any position and fastened with a lock nut, or which may be so made that the cord may be fastened at any one of a number of points around the circumference.

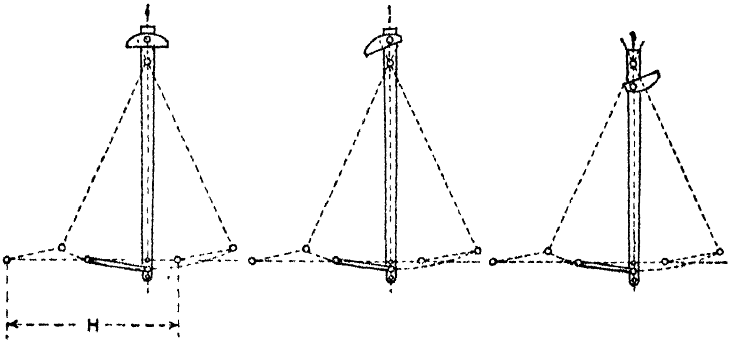


Fig. 430.

Concerning the rest of the construction of pendulum motions, the lower end of the lever may be fastened to the operating pin by an intermediate link, as is indicated in Fig. 431, or the lower end may be slotted, as shown in Fig. 432. In the latter type the standard supporting the lever-pivot must be mounted exactly over the middle of the travel of the pin working in the slot. The former type has the advantage in that it can be mounted to one side of the center of travel, and hence in the case of long-stroke engines this type allows the use of shorter cords. In this case, the following precautions should, however, be observed. The link 1′–1 (see Fig. 433) must be of such a length that the lever 1–6 hangs vertical when the piston is at the middle of the stroke. Further, the length of lever 1–6 should be such that, in connection with the length of link 1–1′, the travel on both sides of the line 1–6 shall be equal, that is, the distance 1–2 should equal distance 1–3. If this is not

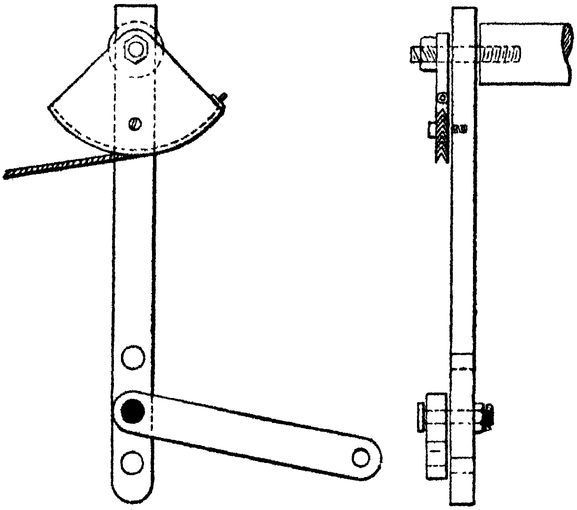


Fig. 431.—Pendulum Reducing Motion with Intermediate Link.

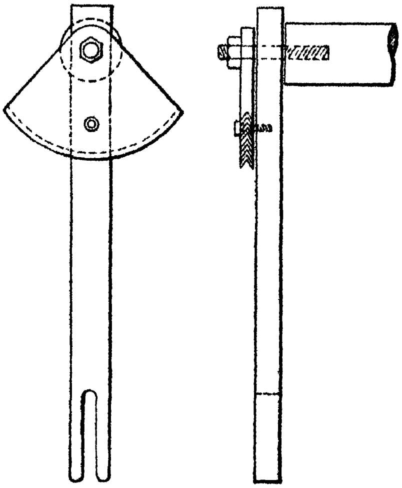


Fig. 432.—Pendulum Reducing Motion with Slotted Main Lever.

approximately observed, large errors may result. Thus, in case of too short a lever 1″–6, the respective travels are 1″–2″ and 1″–3″, which differ in length by the distance 3″–4″, the former being the longer. Or, in case of too long a lever, 1‴–6, the travels are 1‴–2‴ and 1‴–3‴, differing by the distances 4‴–3‴, in this case the latter being the longer.

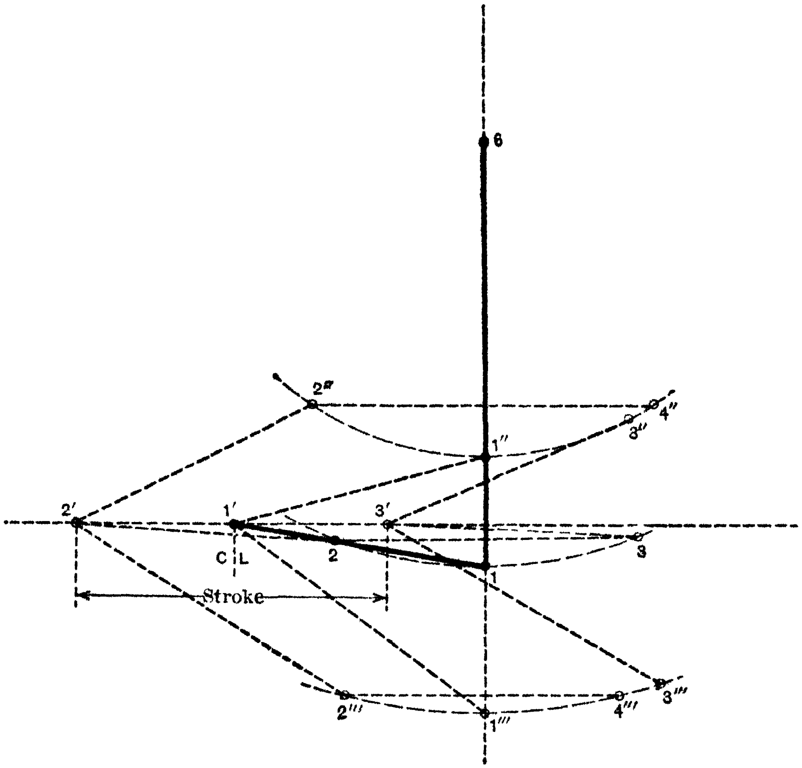


Fig. 433.

In no case, however, will these motions give absolutely correct results, although if the precautions mentioned are observed the errors can be made very small. Loose joints and lost motion should be carefully avoided. In general, the longer the vertical lever, the more accurate the results. For very close work, the length *L*, distance 1–6, in Fig. 433, may be made equal to twice the length of the engine stroke; an ordinary ratio is 1¼ times the length of the stroke. The length of the link, distance 1′–1 in Fig. 433, then follows from the consideration laid down in the previous paragraph.

The statement made at the outset of this discussion, that for absolute accuracy the angles made by the cord should not change at any part of the travel, at once shows that the scheme of simply putting a pin into the end of a rotating shaft, say at *B*, Fig. 434, and using this pin to operate the indicator, gives an incorrect motion. The error is minimized by making the distance from the center *C* of the shaft to the guide pulley *G* as great as possible, with reference to the throw *CB*. For accurate work a small crank mechanism should be substituted for the simple pin. The connecting-rod ratio of this mechanism should be the same as that of the engine, (see Fig. 435.) This type is sometimes modified to the form shown in Fig. 436. This eccentric form has the advantage

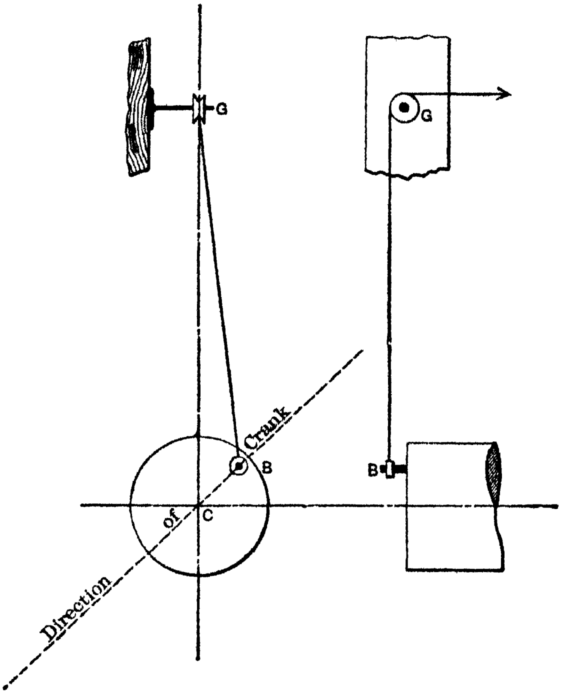


Fig. 434.—Incorrect Reducing Motion.

over the other in that it may be applied to any free length of shaft if the eccentric proper is made in two halves.

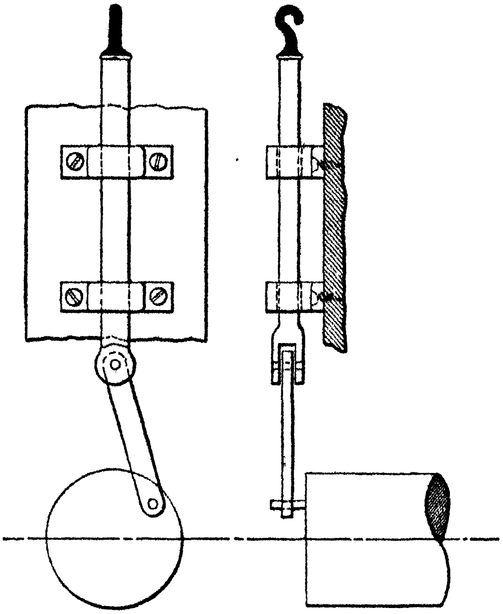


FIG. 435. — CORRECT FORM OF REDUCING MOTION.

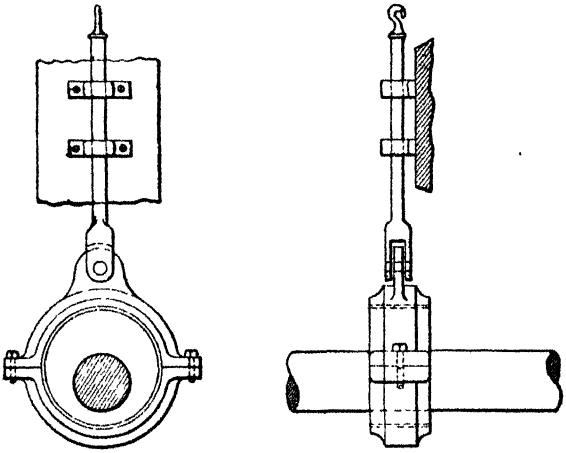


FIG. 436. — REDUCING MOTION ECCENTRIC DRIVE.

The Pantograph. — One form of this instrument, which is sometimes called "lazy tongs," is shown in Fig. 437. It consists of a series of single levers *B* and double levers *A*, joined as shown.

The joints must be accurate and without lost motion, otherwise the instrument may be worse than useless. The hitch-strip *G* is fastened as shown and must be in all positions parallel to the levers *B*. Its distance from the fixed point *D* depends upon the length of card desired. The hitch-pin *F* must be placed in one of the holes in the hitch-strip *G* so that it shall be on a line joining the fixed point *D* to the moving point *C*. Otherwise the motion of *F* will not be parallel to the motion of *C* and the piston travel will not be correctly reduced. The pantograph may be used horizontally, vertically, or in an inclined position. The point *C* is fastened to any reciprocating part of the engine which reproduces the motion of the piston, while the point *D* is rigidly fixed in a stationary support. As for the rest, carry out the following directions:

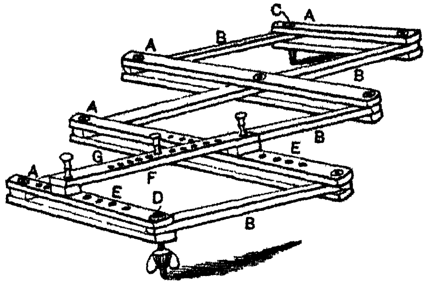


FIG. 437.—PANTOGRAPH REDUCING MOTION.

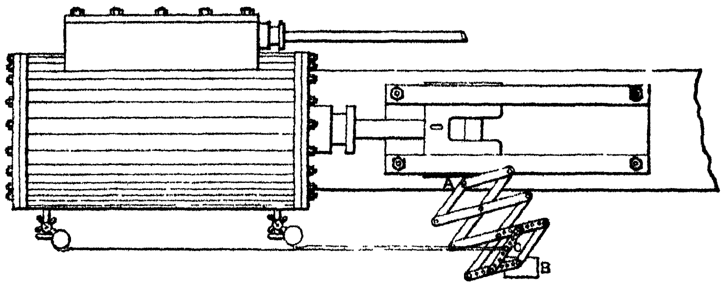


FIG. 438.—PANTOGRAPH APPLIED IN HORIZONTAL POSITION.

1. Place the stationary support so that a line drawn from *D* to *C* will be perpendicular to the line of travel of the member to which *C* is fastened when that member is at the middle of its travel.

2. Move the stationary support in or out so that the pantograph is neither too far extended when at the ends of the stroke nor too nearly closed when at the middle. It is important to see that the pantograph works freely and is not bound in any way by the methods of attaching it.

3. Fasten the hitch-strip according to the length of card desired and locate the hitch-pin in a line from *C* to *D*.

4. See that the indicator-cord leaves the hitch-pin in a line parallel to the crosshead travel. If this cannot be done when leading directly to the indicator, guide pulleys must be used.

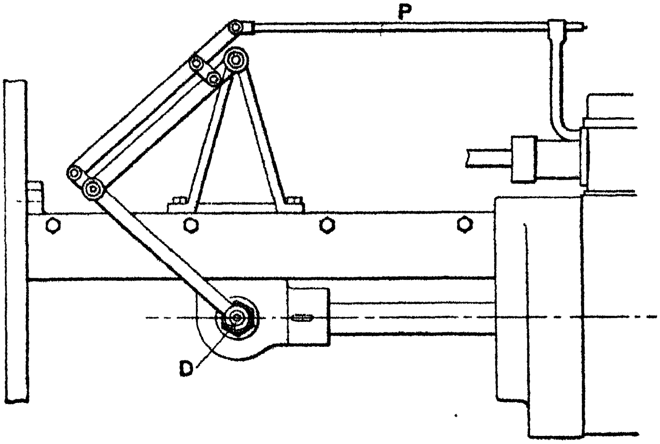


FIG. 439.—PANTOGRAPH APPLIED TO LOCOMOTIVE.

Fig. 438 shows the application of a pantograph of this type in a horizontal position to a horizontal engine. Numerous modifications of the pantograph have been devised and used. One of these is shown in Fig. 439,* as applied to a locomotive. The rod *P* leads to the indicator. Another is shown in Fig. 440, applied to a

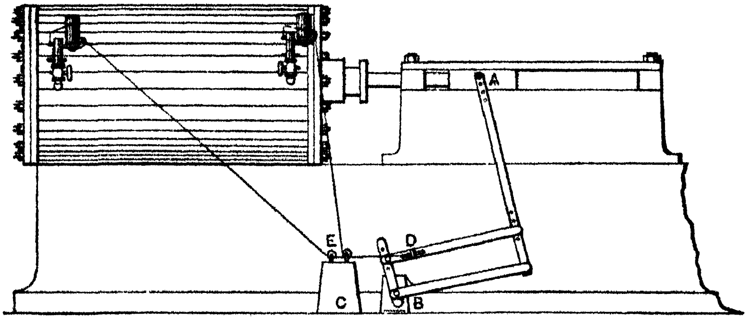


FIG. 440.—MODIFIED FORM OF PANTOGRAPH.

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* P. H. Rosenkranz, Der Indikator, p. 199.

horizontal engine. Note the use of guide pulleys to prevent the change of angles in the strings during operation.

335. **Reducing Wheels.** — These instruments in nearly all cases consist of one large and one small cord drum. The diameter of the latter can usually be changed by slipping sleeves on or off the center arbor. In this way one wheel may be used on engines having widely varying strokes. The action of these wheels is perhaps best explained by means of Figs. 441 and 442, which show a very simple form of reducing wheel. The lever carried by the crosshead unwinds the cord from the large drum on the out-stroke, at the same time winding the cord from the indicator around the small drum. On the return stroke, the strings are kept taut either by the drum-spring of the indicator, or, what is more usual, by a spring included in the construction of the wheel.

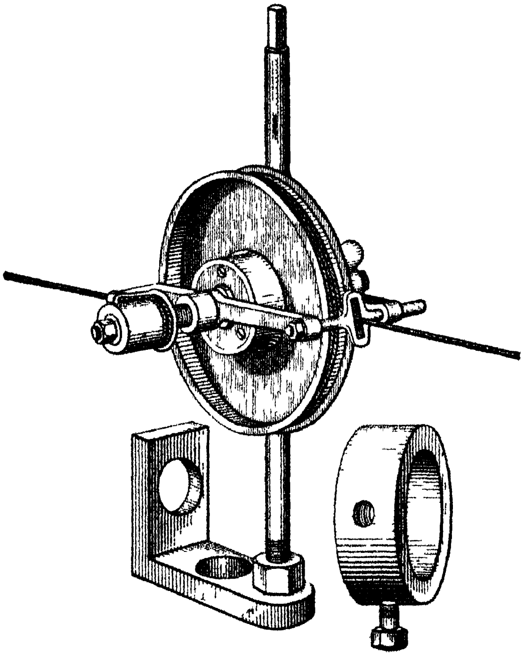


FIG. 441. — REDUCING WHEEL

Other types of reducing wheels are directly connected to the indicator, as the Crosby, Fig. 443, and the American Ideal, Fig. 444.

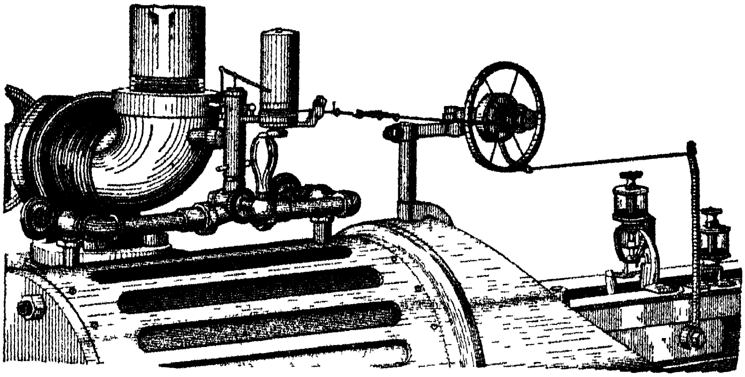


FIG. 442. —REDUCING WHEEL OF FIG. 441 APPLIED.

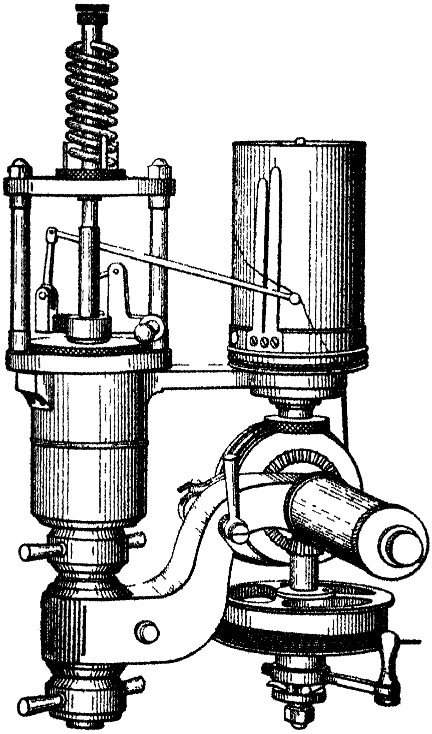


FIG. 443.—CROSBY INDICATOR WITH REDUCING WHEEL.

In each case the string from the large pulley leads to the cross-head or other reciprocating part. Fig. 445 shows still another type, the Houghtaling, as applied to the Tabor indicator.

336. The Indicator-cord and Methods of Connecting up. — The indicator-cord should be as nearly as possible inextensible, since any stretch of the cord causes a corresponding error in the motion of the indicator-drum. As it is nearly impossible to secure a cord that will not stretch, it should be made as short as possible, and a fine wire of steel or iron or of hard-drawn brass should be used if practicable. The indicator-cord supplied by makers of indicators is a braided hard cotton cord, stretching but little under the required stress.

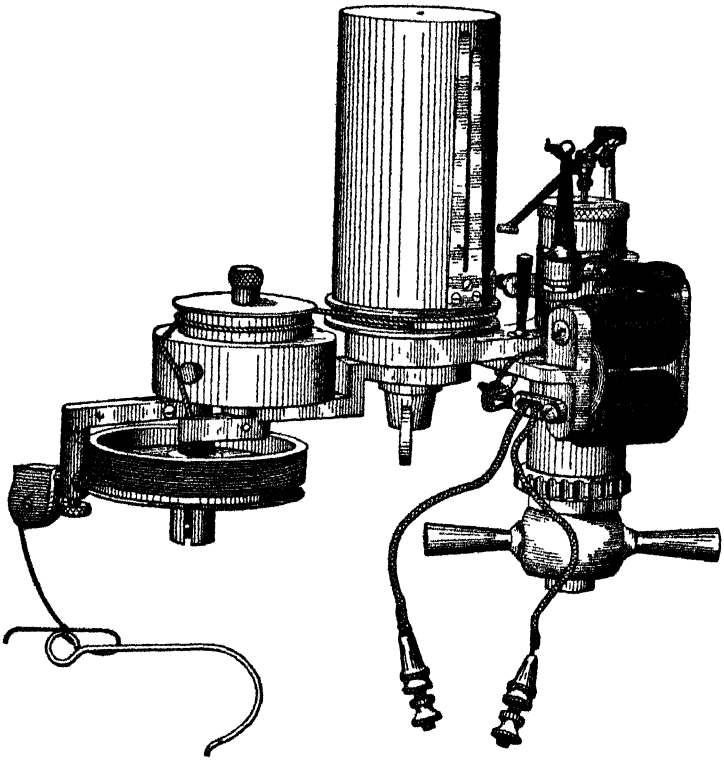


FIG. 444. — THOMPSON INDICATOR WITH AMERICAN IDEAL REDUCING WHEEL.

It pays in many cases to stretch the cord before using by hanging a weight on a suspended length of it and leaving the cord under tension for 10 or 12 hours.

The means for connecting the indicator-cord to the reducing motion, or the cord from the reducing motion to the moving engine member, should be chosen particularly with reference to speed conditions encountered. Where the reducing motion is such that connection can be made on either side, it is generally best to connect on the side having the shortest stroke, that is, on the indicator side, and for obvious reasons. Generally a simple loop in the cord into which a wire hook may be caught serves the purpose. To adjust the length of the cord, an operation that may have to be gone through more than once if the cord is long and has not

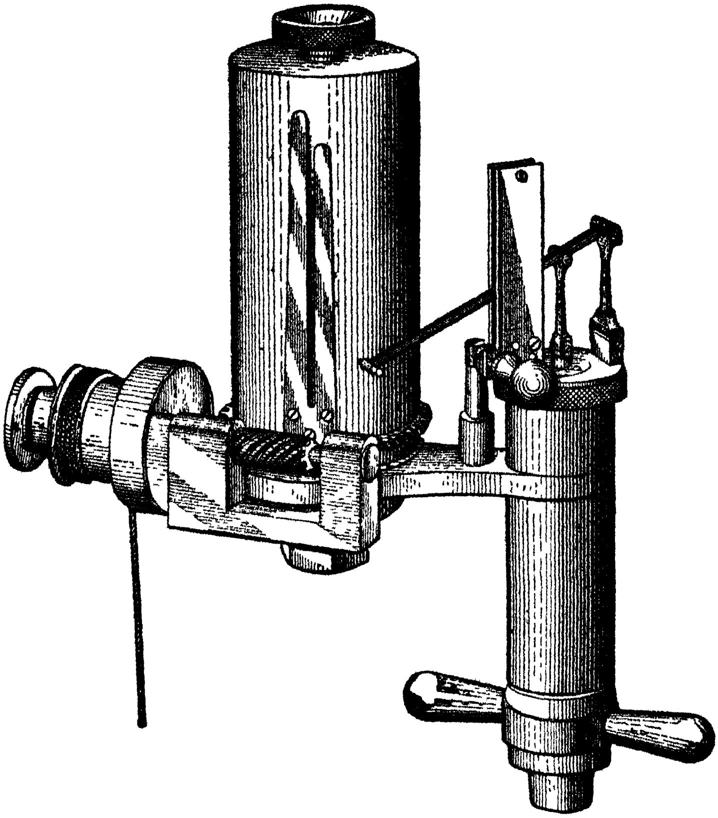


FIG. 445.—TABOR INDICATOR WITH HOUGHTALING REDUCING GEAR.

been previously stretched, a small plate with three holes, used as shown in Fig. 446, comes in very handy. Other means for making quick adjustments of length are shown in Fig. 447.* The main objection to such schemes is that they may be so heavy as to pull the cord out of line.

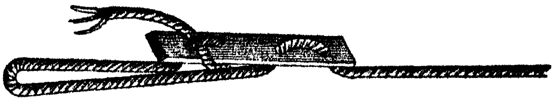


FIG. 446.—ADJUSTING PLATE FOR INDICATOR CORD.

With increase of speed the troubles of taking indicator-cards also increase, and this is especially true of connecting up. The simplest way out of the latter trouble is to leave the indicator connected up after the test is started. That means, however, that the indicator-drum must be fitted with some kind of *detent motion* in

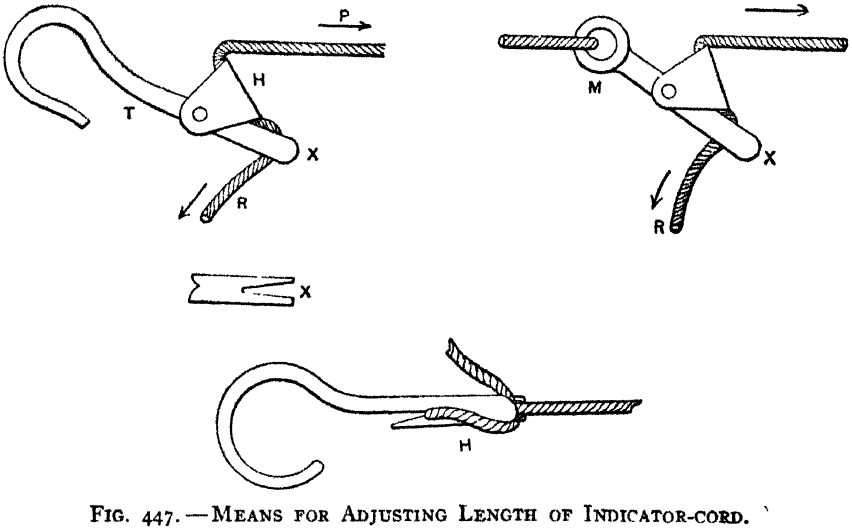


FIG. 447.—MEANS FOR ADJUSTING LENGTH OF INDICATOR-CORD.

order to allow of replacing the cards. There are several types of such motions, and nearly every manufacturer will fit his indicator with one upon request. One of the earliest forms consisted in furnishing the drum with a ratchet near the bottom into which a

* From Rosenkrang, Der Indikator.

pawl could be pushed when the reducing motion had pulled the cord out to the limit. This arrested the back-motion of the drum, but left the string slack, which was likely to cause whipping and snarling up. A better scheme, and one often used now, is to make the drum-support in two parts, which can be connected and disconnected at will by some means or other. This allows the drum-support with its spring to continue, keeping the string taut, but puts the paper drum itself at rest. Detent motions are very useful under high speeds in the case of pendulums, pantographs, and other motions which cannot well be set at rest during the interval between cards. In the case of reducing wheels, they do not help matters much, as it is usually not desirable to keep such wheels going all the time. Hence the connection must often be made on the engine side, and the matter is often difficult on account of high speed or long stroke, or both. A scheme that the writer has sometimes used in cases where the stroke is not too long is to lead the cord from its point of fastening on the moving engine member closely past the reducing wheel and beyond the wheel to a stationary point of support, inserting a length of helical spring just ahead of the latter point. This keeps the string taut and in motion all the time. Then ahead of the reducing wheel insert into the string a closed link of light but stiff steel wire. Although the latter will then move back and forth continuously the full engine stroke, it is not particularly difficult to hook into it the wire hook carried on the end of the reducing wheel string. Another scheme is to make a special snap hook which can be hooked in very readily with a little practice. The hook is tied directly to the cord from the wheel and is constructed as shown in Fig. 448, from which the operation should be clear.

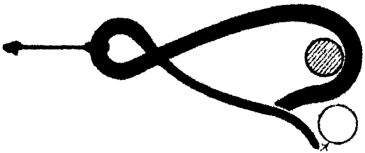


FIG. 448. — SNAP HOOK FOR CONNECTING INDICATOR-CORD.

337. Electrical Attachments for Taking a Number of Cards at the Same Time. — It is sometimes desirable to obtain cards from a number of indicators simultaneously. This can be done by means of electromagnetic attachments, the usual principle of which is to mount on the drum support a small electromagnet, the armature for which is fastened to the collar carrying the pencil motion. A light spring holds the armature away from the poles when the circuit is open, thus keeping the pencil away from the paper. The electromagnets and all the indicators, are connected so that the closing of a single switch takes all of the diagrams at once. One attachment of this type is shown in Fig. 444.

338. Care of the Indicator. — The indicator is a delicate instrument, and its accuracy is liable to be impaired by rough usage. It must be handled with care, kept clean and bright; its journals must be kept oiled with suitable oil. It must be kept in adjustment. Before using it, take it apart, clean and oil it. Try each part separately. See if it works smoothly; if so, put it together without the spring. Lift the pencil-lever, and let it fall; if perfectly free, insert the spring as explained, and see that there is no lost motion; oil the piston with cylinder-oil, and all the bearings with nut- or best sperm-oil. If the oil from the engine gums the indicator, always take it off and clean it. After using it remove the spring, dry it and all parts of the indicator, then wipe off with oily waste. Fasten the indicator in its box, in which it will go as a rule only one way, but it requires no pounding to get it properly into place; carefully close the box to protect the contents from dust.

339. Directions for Taking Indicator-diagrams.

1. Provide a perfect reducing-motion, and make arrangements so that the indicator-drum can be stopped or started at full speed of the engine.

2. Clean and oil the indicator, and attach it to the engine as previously explained. Insert proper spring; oil piston with cylinder-oil.

3. Put proper tension on the drum-spring; see that the pencil-point is sharp and will draw a fine line.

4. Connect the indicator-cord to the reducing-motion; turn the engine over and adjust the cord so that the indicator-drum has the proper movement and does not hit the stops.

5. Put the paper on the drum; turn on steam, allow it to blow through the relief-hole in the side of the cock; then admit steam to the indicator-cylinder, close the indicator-cock, start the drum in

motion, and draw the atmospheric line with engine and drum in motion; open the cock, press the pencil lightly, and take the diagram; close the cock and draw a second atmospheric line. Do not try to obtain a heavy diagram, as all pressure on the card increases the indicator friction and causes more or less error. Take as *light a card* as can be seen; brass point and metallic paper are to be used when especially fine diagrams are required.

When the load is varying, and the *average* horse power is required, it is better to allow the pencil to remain during a number of revolutions, and to take the mean effective pressure from the several diagrams drawn.

Remove card after diagram has been taken, and on the back of card make note of the following particulars, as far as conveniently obtainable:

No . . . .
Time . . . .
Date. . . . .
Make of Engine. . . . .
Built by. . . . .
Diam. of Cyl. . . . .
Length of Stroke. . . . .
R p m. . . . .
Scale of Spring. . . . .
PRESSURES:
At Boiler . . . .
At Throttle. . . . .
Vacuum . . . .
Remarks. . . . .

6. *After a sufficient number of diagrams has been taken,* remove the piston, spring, etc., from the indicator while it is still upon the cylinder; allow the steam to blow for a moment through the indicator-cylinder, and then turn attention to the piston, spring, and all movable parts, which must be thoroughly wiped, oiled, and cleaned. *Particular attention* should be paid to the springs, as their accuracy will be impaired if they are allowed to rust; and great care should be exercised that no gritty substance be introduced to cut the cylinder or mar the piston. Be careful never to bend the steel bars or rods.

The above directions apply in general also to all types of engines.

340. **The Optical Indicator.** — Many attempts have been made to modify the piston indicator in such a way as to overcome the difficulties arising from inertia effects, particularly for high-speed

work. The only successful instruments which have resulted are those known as optical indicators. In all of these the movement is very small and the parts moved are very light.

The general scheme is to have a small mirror given two simultaneous motions at right angles to each other. Of these, one is proportional to the engine piston motion and the other proportional to the pressure in the engine cylinder. A beam of light reflected from the mirror traces a diagram on a piece of plate glass or a sensitized plate or paper.

These indicators can be used successfully up to very high speeds and pressures, but they labor under the disadvantages that they are even more difficult to properly handle than are the piston indicators, and that if the diagram is to be preserved it must be photographed. This latter is a delicate and time-consuming as well as comparatively expensive process.

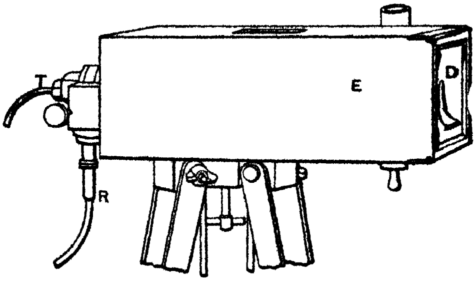


FIG. 449. — MANOGRAPH.

The two forms of optical indicators described below are good examples of the types produced and are probably as accurate as any.

One form of this instrument is made by J. Carpentier of Paris, and is called the *Manograph.* Another form similar to it is made by the Elsässische Elektricitäts-Werke, Strassburg.

A perspective view and section of the manograph is shown in Figs. 449 and 450. A small mirror is located at *A*, Fig. 450, in the back part of the camera. It is deflected in one direction by a small crank operated in unison with the engine piston by the revolving shaft *P*, to which it is connected by the flexible shaft *R*, Fig. 449; it is deflected in a direction at right angles against the resistance of a spring by the pressure from the engine cylinder acting through a pipe *T* upon a diaphragm directly back of the mirror.

The mirror is illuminated by light from a lamp at *G*, which is reflected by the prism shown at *H*. The indicator-diagram is traced on the screen *D* by the ray of light, and may be photographed by the use of a sensitive plate. This apparatus has been successfully used to take indicator-diagrams of gas engines when moving at the rate of 2000 revolutions per minute.

In its original form this indicator was a very inaccurate instrument, principally because of the method of connecting the diaphragm chamber to the engine cylinder and because of the method of imparting the motions supposed to be proportional to that of the piston.

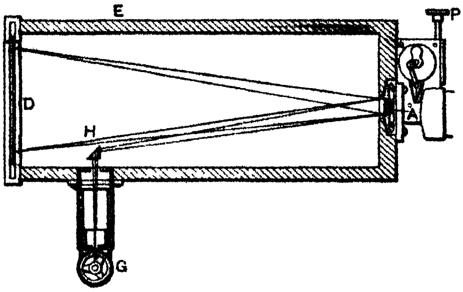
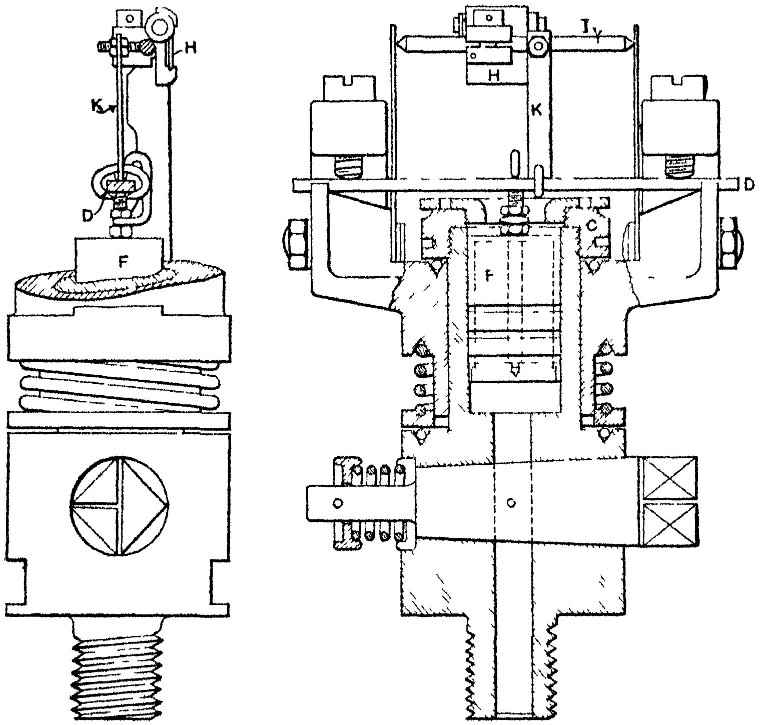


FIG. 450.—MANOGRAPH SECTION.

The pressure was transmitted by means of a long copper pipe of very small bore, and the pressure actually existing in the diaphragm chamber might have been anything less than that in the cylinder of the engine. In recent work it has become the custom to fasten the camera box and with it the diaphragm chamber directly to the indicator connection on the engine cylinder.

The motion transmitted to the mirror as proportional and in phase with that of the engine was carried by a long flexible shaft, and it was impossible to tell whether the proper phase relation existed or not. If set correctly with everything at rest it certainly was not correct when in motion at high speed. This difficulty is now generally overcome, in as far as that is possible, by the use of positive linkages or similar connections, so arranged that the proper phase can be very nearly maintained.

The indicator labors under the further disadvantage, common to all diaphragm instruments, that the deflections become smaller for equal increments of pressure the higher that pressure. This makes it a difficult matter to correctly evaluate the area of the diagram, though of course it does not impair the usefulness of the instrument as a means of indicating the correct or incorrect setting of the valves.



FIGS. 451 AND 452.—HOPKINSON OPTICAL INDICATOR.

The *Hopkinson Optical Indicator* was designed later than most of the others at present in use, and after the designer had become familiar with the difficulties met with in the use of the type last described. It is shown in Figs. 451 and 452.*

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* *Power*, Jan. 19, 1909.

It is a piston type of indicator in a way, but as the extreme motion of the piston *F* is only $\frac{1}{40}$ of an inch the inertia effects are practically negligible. The motion of the piston is resisted by the flat spring *D*, and the deflection of the spring is transmitted to a small mirror by the flexible link *K*. This mirror rocks on the horizontal axle *I* by an amount proportional to the deflection of the spring. The whole upper part of the indicator is rotated about the vertical axis of the instrument to obtain the motion of the mirror which is to be proportional to that of the piston. As the necessary rotation of this head is but $3\frac{1}{2}$ degrees for a two-inch card, the inertia effects are again very small.

The instrument is fastened directly to the engine cylinder, as is the ordinary form of indicator, thus doing away with long restricted pipes, and it is positively driven by various forms of reducing motion, adapted to the high speeds for which the indicator is intended.

By means of two springs of different strengths, and of three pistons of different areas, a very wide range of pressures can be covered with one instrument.

As in other forms of optical indicator, the diagram may be thrown upon a plate of ground glass for purposes of observation only, or may be photographed for record. The optical arrangements are such that the spot of light tracing the diagram can be focused to a very small point, thus preventing the blurring of the photographed lines, a fault which is very common in some of the other types of optical indicator.

CHAPTER XVI.

THE INDICATOR-DIAGRAM.

I. STEAM-ENGINE DIAGRAMS.

341. **Definitions.** — The method of obtaining diagrams by means of the engine indicator has already been explained in the previous chapter.

In the diagram the ordinates correspond to the pressures per square inch acting on the piston, the abscissæ to the travel of the piston. During a complete revolution of a steam engine occur *four phases of valve-motion* which are shown on the indicator-diagram, viz.: *admission*, *CDE*, Fig. 453, when the valve is open and the steam is passing into the cylinder; *expansion*, *EF*, when steam is neither admitted nor released and acts by its expansive force to move the piston; *exhaust*, *FGH*, when the exhaust port is open so

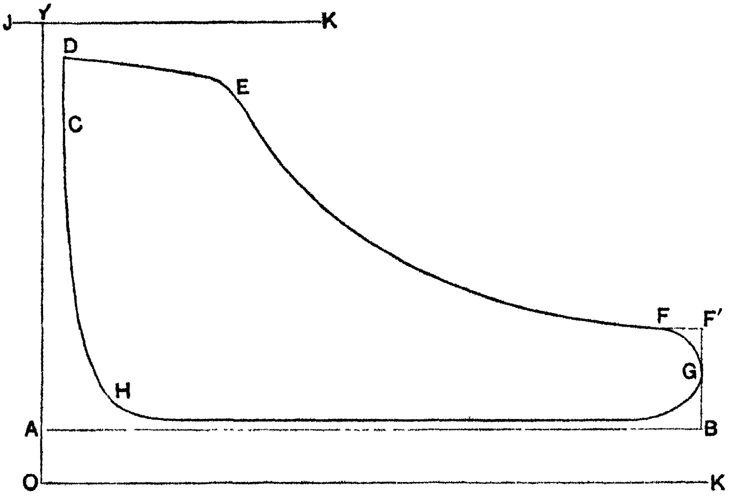

FIG. 453. — DIAGRAM FROM NON-CONDENSING STEAM ENGINE.

that steam is escaping from the cylinder; and *compression*, HC, when all the ports are closed and the steam remaining in the cylinder acts to bring the piston to rest.

The Atmospheric Line, AB, is a line drawn by the pencil of the indicator when the connections with the engine are closed and both sides of the piston are open to the atmosphere. This line represents on the diagram the pressure of the atmosphere, or zero gauge-pressure.

The Vacuum Line, OK, is a reference-line drawn a distance corresponding to the barometer-pressure (usually about 14.7 pounds) by scale below the atmospheric line. It represents a perfect vacuum, or absence of all pressure.

The Clearance Line, OY, is a reference-line drawn at a distance from the end of the diagram equal to the same per cent of the diagram length as the clearance or volume not swept through by the piston is of the piston-displacement. The distance between the clearance line and the end of the diagram represents the volume of the clearance of the ports and passages at the end of the cylinder.

The Line of Boiler-pressure, JK, is drawn parallel to the atmospheric line, and at a distance from it by scale equal to the boiler-pressure shown by the gauge. The difference in pounds between it and DE shows the loss of pressure due to the steam pipe and the ports and passages in the engine.

The Admission Line, CD, shows the rise of pressure due to the admission of steam to the cylinder by opening the steam valve. If the steam is admitted quickly when the engine is about on the dead-center, this line will be nearly vertical.

The Point of Admission, C, indicates the pressure when the admission of steam begins at the opening of the valve.

The Steam Line, DE, is drawn when the steam valve is open and steam is being admitted to the cylinder.

The Point of Cut-off, E, is the point where the admission of steam is stopped by the closing of the valve. It is difficult to determine the exact point at which the cut-off takes place. It is usually located where the outline of the diagram changes its curvature from convex to concave. It is most accurately determined by extending the expansion line and steam line so that they meet at a point. (See also Chapter XVIII.)

The Expansion Curve, EF, shows the fall in pressure as the steam in the cylinder expands doing work.

The Point of Release, F, shows when the exhaust valve opens.

The Exhaust Line, FG, represents the change in pressure that takes place when the exhaust valve opens.

The Back-pressure Line, GH, shows the pressure against which the piston acts during its return stroke. On diagrams taken from non-condensing engines it is either coincident with or above the atmospheric line. On cards taken from condensing engines it is found below the atmospheric line, and at a distance greater or less according to the vacuum obtained in the cylinder.

The Point of Exhaust Closure, H, is the point where the exhaust valve closes and compression begins. It cannot be located very definitely, as the first slight change in pressure is due to the gradual closing of the valve.

The Compression Curve, HC, shows the rise in pressure due to the compression of the steam remaining in the cylinder after the exhaust-valve has closed.

The Initial Pressure is the pressure acting on the piston at the beginning of the stroke.

The Terminal Pressure is the pressure that would exist at the end of the stroke if the steam had not been already released. It is found by continuing the expansion curve to the point F', Fig. 453.

Admission Pressure is the pressure acting on the piston at end of compression, and is usually less than initial pressure.

Compression Pressure is the pressure acting on the piston at beginning of compression.

Cut-off Pressure is the pressure acting on the piston at beginning of expansion.

Release Pressure is the pressure acting on the piston at end of expansion.

Mean Forward Pressure is the average height of that part of the diagram traced on the forward stroke.

Mean Back Pressure is the average height of that part traced on the return stroke.

Any of these pressures may be measured either above the atmospheric line or above the vacuum line. In the former case they are termed "gauge pressures," in the latter, "absolute pressures."

Mean Effective Pressure (M.E.P.) is the difference between mean forward and mean back pressure during a forward and return stroke. It is the length of the mean ordinate intercepted between the top and bottom lines of the diagram multiplied by the scale of the diagram. It is obtained without regard to atmospheric or vacuum lines.

Ratio of Expansion is the ratio of the total cylinder volume to the total volume at cut-off. In computations for this quantity the volume of clearance is therefore taken into account. Ratio of expansion is denoted by r. (See also p. 748.)

The volumes may be expressed as proportional to linear feet, with an additional length equal to the per cent of clearance, since the area of the cylinder is constant.

The practice of engineers seems to differ with regard to the method of determining cut-off and the ratio of expansion. In this connection see the discussion on the analysis of indicator diagrams in Rule XX of the Steam Engine Testing Code, Chapter XVIII. This takes up the cases of single and multicylinder, condensing and non-condensing engines.

Wire-drawing is the fall of pressure between the boiler and cylinder, indicated by the difference between lines *JK* and *DE*.

342. **Measurements from the Diagrams.** — The diagrams taken are on a small scale; they are often irregular, and the boundary lines are frequently obscure, so that the measurement must be made with great care.

The diagrams may be taken from each end of the cylinder on a separate card, as shown in Fig. 453, or by the use of the three-way cock (see Article 333), in which case the two diagrams will be drawn on the same card as shown in Fig. 454. In the latter case each diagram is to be considered separately; that is, the area of each diagram, as *CDEBFC* and *GHIJKG*, is to be determined as though on a separate card. The object of diagram measurements is principally to obtain the mean effective pressure (M.E.P.).

Two methods are practiced.

First, the *method of ordinates.* In this case the length of the diagram is divided into ten equal spaces, and ordinates are erected from the center of each space. The sum of the length of these various ordinates divided by the number gives the mean ordinate. This multiplied by the scale of the indicator spring gives the mean effective pressure. The sum of the ordinates is expeditiously obtained by successively transferring the length of each ordinate to a strip of paper and measuring its total length.

Secondly, *determination of area by means of the planimeter.* This method gives the mean ordinate much more accurately and quickly than the method of ordinates. The various planimeters are fully described, pages 21 to 42.

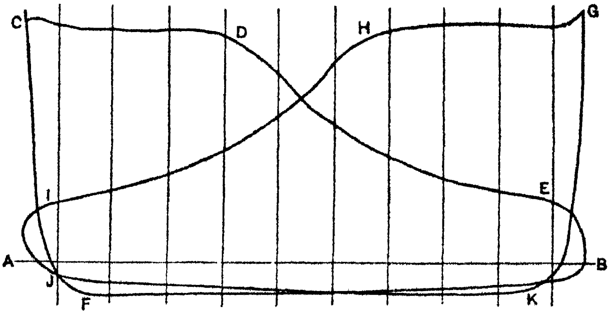

FIG. 454. — DIAGRAM FROM CONDENSING ENGINE ON ONE CARD.

With any planimeter the area of the diagram can be obtained, in which case the mean ordinate is to be found by dividing by the length of the diagram. Several of the planimeters give the value of the mean ordinate directly.

In some instances the indicator-diagram has a loop, as in Fig. 455, caused by expanding below the back-pressure line; in this case the ordinates to the loop are negative and should be subtracted from the lengths of the ordinates above. In case of measurement by the planimeter, if the tracing-point be made to follow the expansion-line in the order it was drawn by the indicator-pencil, the part within the loop will be circumscribed by a reverse motion, and will be deducted automatically by the instrument, so that the reading of the planimeter will be the result sought. Concerning scales of springs, by which the mean ordinate is multiplied to get the *mean effective* pressure, see pages 608 to 612, Chapter XV.

343. **Indicated Horse Power.** — Indicated horse power is the horse power computed from the indicator-diagram, being obtained by the product of M.E.P. (p), length of stroke in feet (l), area of piston in square inches (a), and number of revolutions (n), as represented in the formula $plan \div 33,000$. (See also Art. 317.) In this computation the area on the crank side of the piston is to be corrected for area of piston-rod, and the two ends of the cylinders computed as separate engines. Further, in this computation it will not in general answer to multiply the average M.E.P. of a number of cards by the length of stroke and by the average of the number of revolutions, but each card must be subjected to a separate computation and the results averaged. This can be readily done for each engine by computing a table made up of the products of the average value of n by length of stroke l and area of piston a, and for different values of M.E.P. Take from this table the values corresponding to the given M.E.P., and increase or diminish this as required by the per cent of change of speed from the average. A very convenient table for this purpose, entitled "Horse Power per Pound Mean Pressure," is given in the Appendix to this work, arranged with reference to diameter of cylinder in inches and piston-speed in feet per minute. *Piston-speed* in feet per minute is the product of length of stroke in feet by revolutions per minute = ln.

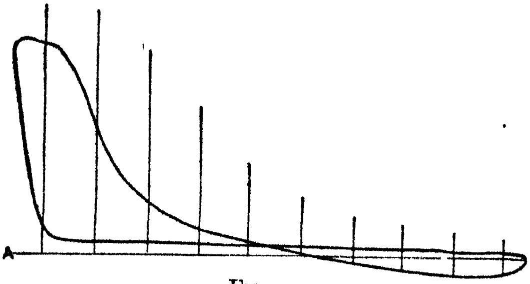

FIG. 455.

344. **Form of the Indicator-diagram.** — The form of the indicator-diagram has been carefully worked out for the ideal case by Rankine

and Cotterell.* In the ideal case the steam works in a non-conducting cylinder, and all loss of heat is due to transformation into work, the expansion in such a case being adiabatic. In the actual case the problem is much more complicated, since a large portion of the heat is utilized in heating the cylinder, and is returned to the steam at or near the time of exhaust, doing little work. It is found, however, in the best engines working with quick-acting valve-gear, that the steam and back-pressure lines are straight and parallel to the atmospheric line, and that the expansion and compression lines are very nearly hyperbolæ, asymptotic to the clearance line and to the vacuum line.

If we denote by p the pressure measured from the vacuum line, and by v the volume corresponding to a distance measured from the clearance line, so that pv shall be the coördinates of any point, we shall have as characteristic of the hyperbola

$$pv = \text{constant}. \tag{1}$$

This is the same as Boyle's law for the expansion of non-condensible gases, since, according to that law, the pressure varies inversely as the volume.

Rankine found by examination of a great many actual cases that the expression $pv^{\frac{9}{8}}$ = constant agrees very nearly with the ideal case of adiabatic (isentropic) expansion.† The variation from the ideal expansion line in any given case may be considerable, and the hyperbola drawn from the same origin is considered as good a reference line as any that can conveniently be used. The student should therefore become familiar with the best methods of constructing it.

345. Methods of Drawing an Hyperbola. — The methods of drawing an hyperbola, the clearance and vacuum lines being given, are as follows:

First Method. (See Fig. 456.) — CB, the clearance line, and CD, the vacuum line, being given, draw a line parallel to the atmospheric line through B; find by producing expansion line the point of cut-off, *c*. Draw a series of radiating lines from the point *C* to the points *E*, *F*, *G*, *H*, and *A*, taken at random, and a line *cb* intersecting these lines, drawn from *c* parallel to *BC*. From the points of the intersection of *cb* with these radiating lines draw horizontal lines to meet vertical lines drawn from the points *E*, *F*, *G*, *H*, and *A*; the intersections of these lines at *e*, *f*, *g*, *h*, and *a* are points in the hyperbola passing through the point *c*. If it is desired to produce the hyperbola from *a* upward, the same method is used, but

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* Steam Engine, by James H. Cotterell.

† For more detailed information see under "Adiabatic (Isentropic) Change for Steam" in Chapter XI.

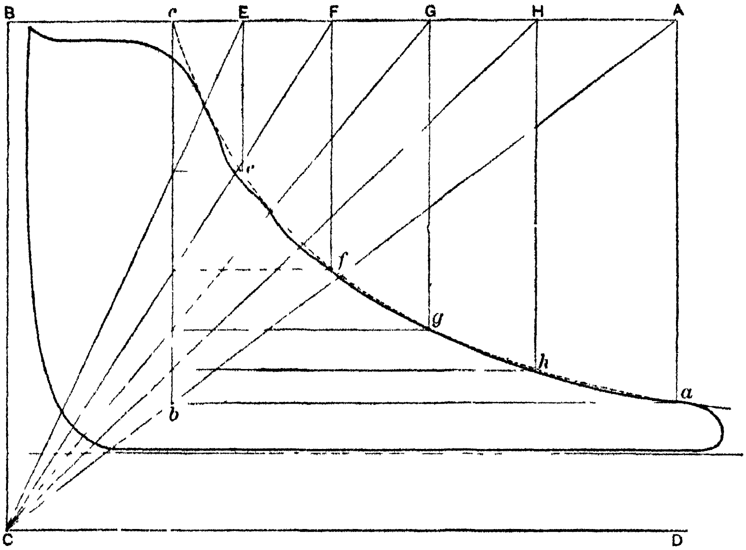

FIG. 456 — METHOD OF DRAWING HYPERBOLA.

the line *AB* is drawn through the point *a*, and the vertical lines are extended above *AB* instead of below.

Second Method. (See Fig. 457.) — The hyperbola may be drawn by a method founded on the principle that the intercepts made by a straight line intersecting an hyperbola and its asymptotes are equal. Thus if *abcd* represents an hyperbola, *BC* and *CD* its asymptotes, then the intercepts *aa′* and *bb′* made by the straight line *a′b′* are equal.

To draw the hyperbola: Beginning at any point, as *a*, draw the straight line *a′b′*, and lay off from the line *CD*, *b′b* equal to *a′a*; then will *b* be one point on the hyperbola. Draw a similar line *c′d′*

through *b*, making *d'c* equal *c'b*; then will *c* be another point on the hyperbola. This process can be repeated until a suitable number of points is found; the hyperbola is to be drawn through these points. A similar method can be used to draw the hyperbola *EF*.

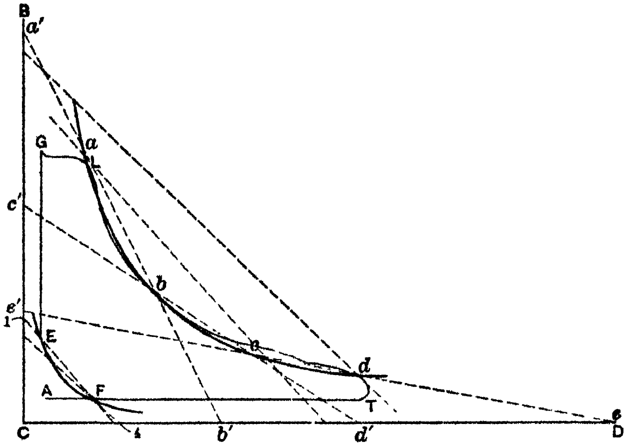

FIG. 457. — METHOD OF DRAWING HYPERBOLA.

346. Construction of Saturation and Isentropic* Curves for Steam.

— The *Saturation Curve* is the curve which results if the specific volumes of dry and saturated steam are plotted against corresponding absolute pressures. No doubt the easiest way to construct such a curve is to take the volumes from the steam-tables corresponding to given pressures and set them off along the volume axis; lay off the corresponding pressures as ordinates; then a curve drawn through the extremities of the ordinates will be the saturation curve, which does not differ greatly from an hyperbola.

It has been stated that the saturation curve for steam follows the general law,

$$pv^{\frac{17}{16}} = pv^{1.064} = \text{constant}. \qquad (2)$$

As a matter of fact, this exponent changes with the pressure range as computations by means of a steam-table will show. The curve obtained by the use of $n = 1.064$ is, however, probably close enough to the real curve to serve all practical purposes.

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* This curve is in general practice called an adiabatic curve. See note, p. 344.

The equation of the isentropic curve for saturated steam (see equation (59), p. 346) is, for all qualities between .7 and 1.0,

$$pv^{1.035+.1x} = \text{constant}. \tag{3}$$

For initially dry and saturated steam ($x = 1.0$) this reduces to

$$pv^{1.135} = \text{constant}. \tag{4}$$

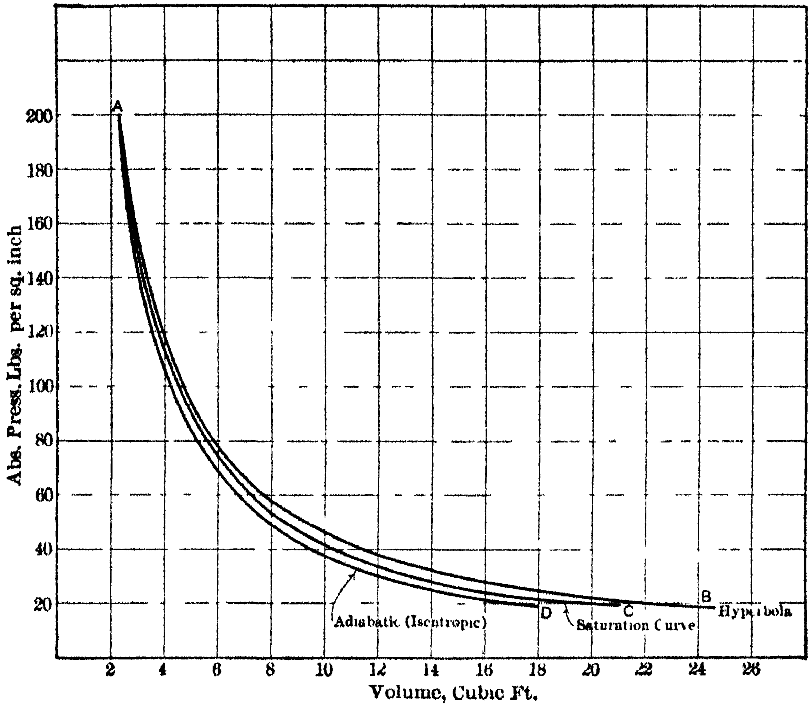


FIG 458

The exponent of the equilateral hyperbola is $n = 1$, and this curve, together with the saturation curve and the isentropic curve for initially dry steam, is plotted in Fig. 458 to show the mutual relation. AB is the hyperbola, AC the saturation curve, and AD the isentropic line.

It is to be noticed that the saturation curve corresponds to a uniform quality of steam, the isentropic curve to a condition in which the moisture is increasing, and the hyperbolic curve to a condition in which the moisture is decreasing.

347. Clearance Determined from the Diagram. — The clearance is usually to be determined by actual measurement of the volume of the spaces not swept through by the piston, and comparing this result with the volume of piston-displacement. Since the expansion and compression lines of the diagram are nearly hyperbolæ, the position of clearance line can be determined by a method nearly the reverse of that used in constructing an hyperbola (Article 345).

In this case proceed as follows: Lay off the vacuum line *CD* (Fig. 459) parallel to the atmospheric line *FT*, and at a distance corresponding to the atmospheric pressure. The position of the clearance line can be determined by two methods, corresponding to those used in drawing the hyperbola. *First method:* Take two points, *a* and *b* in the expansion curve and *c* and *d* in the compression line, and draw horizontal and vertical lines through these points, forming rectangles *aa'bb'* and *cc'dd'*. Draw the diagonal of either rectangle, as *a'b'*, to meet the vacuum line *CD*; the point of intersection *C* will be a point in the clearance line *CB*, and the clearance will equal *CN* ÷ *FT*. *Second method:* Draw a straight line through either curve, as *mn* through the compression curve or *ef* through the expansion curve, and extend it in both directions. On the line

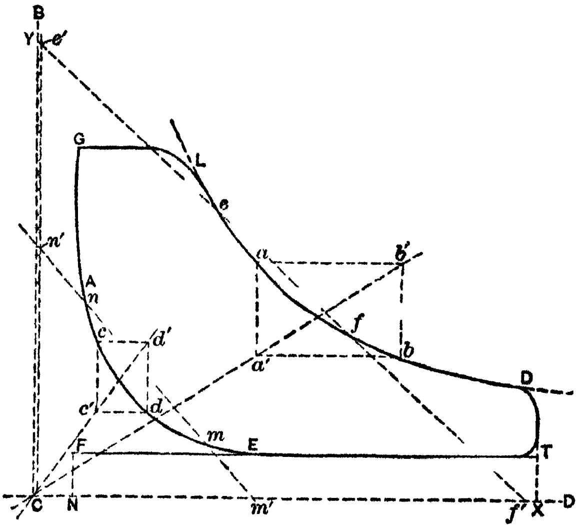


FIG. 459.

m'n' lay off *nn'* equal to *mm'*, or on the line *e'f'* lay off *ee'* equal to *ff'*; then will either of the points *e'* or *n'* be in the clearance line, and the line drawn perpendicular to the vacuum line through one of these points should pass through the other. In an engine working with much compression the clearance will be given more accurately from the compression curve than from the expansion curve, since it is more nearly an hyperbola.

348. **Weight of Steam Accounted for by the Indicator-diagram. The Diagram Water-rate.** — The diagram shows by direct measurement the pressure and volume at any point in the stroke of the piston; the weight of steam per cubic foot for any given pressure may be taken directly from a steam-table. The method, then, of finding the weight of steam for any point in the stroke is to find the volume in cubic feet, including the clearance and piston-displacement to the given point, which must be taken at cut-off or later, and multiply this by the weight per cubic foot corresponding to the pressure at the given point as measured on the diagram. This will give the weight of steam in the cylinder accounted for by the indicator-diagram, per stroke. In an engine working with compression, the weight of steam filling the clearance-space is not exhausted; this weight, computed for a volume equal to clearance and with weight per cubic foot corresponding to compression pressure, should be subtracted from the above. The result may be reduced to pounds of steam per I.H.P. per hour, by multiplying by the number of strokes made per hour and dividing by the I.H.P. developed. It should be noted in this computation that the steam caught in clearance is assumed dry at the end of compression. An alternative method, often used for computing the weight of steam caught in clearance, is to assume the steam dry at the point of exhaust closure and to determine the weight from the cylinder volume up to that point and from the back pressure.

The method of computing would then be: Find, from a steam-table, the weight of steam per cubic foot corresponding to the absolute pressure at the given point, multiply this by the corresponding volume in cubic feet, including clearance, and this by the number of strokes per hour. Correct this for the steam imprisoned in the clearance-space. Divide this by the horse power developed and we shall have the consumption in pounds of dry steam per I.H.P. as shown by the diagram. Thus let

A = area of piston in square feet;

a = “ “ “ “ “ inches;

N = number of strokes per hour;

n = “ “ “ “ minute;

w = weight of cubic foot of steam for the pressure at the point under consideration;

w' = weight of cubic foot of steam for the pressure at end of compression;

l = total length of stroke in feet;

l_a = length of stroke in feet to point under consideration;

c = per cent of clearance;

$l' = l_a + cl$ = equivalent length of stroke to point under consideration (including clearance);

$b = \frac{l'}{l}$.

Then the total volume up to the point under consideration will be $= (l_a + cl)A = l'A$ cu. ft., and the corresponding weight of dry steam is $l'Aw$ pounds. The volume of steam caught in the clearance space is clA cu. ft., and its weight will be $clAw'$ pounds. Hence the net weight of steam supplied, as shown by the indicator, will be $(l'Aw - clAw')$ pounds per stroke. From this it follows that the steam consumption from the diagram in pounds per I.H.P. hour, or the diagram water-rate, is

$$\begin{aligned}
S &= \frac{N}{\text{I.H.P.}}[l'Aw - clAw'] \text{ pounds} \\
&= \frac{NAl}{\text{I.H.P.}}\left[\frac{l'}{l}w - cw'\right]\text{pounds} \\
&= \frac{NAl}{\text{I.H.P.}}(bw - cw') \text{ pounds} \\
&= \frac{60\,n \cdot \frac{a}{144} \cdot l\,(bw - cw')}{\frac{plan}{33{,}000}} \text{pounds} \\
&= \frac{13{,}750\,(bw - cw')}{p} \text{ pounds.} \qquad (5)
\end{aligned}$$

Example. — Compute the steam consumption as shown at E and at release point F, in the diagram of Fig. 453.

The absolute pressures shown by the diagram are as follows:

At cut-off, 97 pounds; at release, 37 pounds; at end of compression, 60 pounds. The mean effective pressure, $p = 50$ pounds; length of stroke, $l = 3$ feet; distance up to cut-off, $l_a = .75$ foot; distance up to release, $l_a = 2.78$ feet.

Per cent of clearance, $c = 3.2$ per cent; $l' = l_a + cl = .75 + .032 \times 3 = .845$ foot at cut-off, and $2.78 + .032 \times 3 = 2.876$ feet at release. Hence $b = \frac{l'}{l} = \frac{.845}{3} = .282$ at cut-off, and $= \frac{2.876}{3} = .958$ at release.

w at cut-off $= .2193$ pound per cubic foot; at release $= .0886$ pound per cubic foot, and w' at end of compression $= .1394$ pound per cubic foot.

Steam consumption at cut-off:

$$S = \frac{13,750\,(.282 \times .2193 - .032 \times .1394)}{50}$$
$$= 15.78 \text{ lbs. per I.H.P. hr.}$$

Steam consumption at release:

$$S = \frac{13,750\,(.958 \times .0886 - .032 \times .1394)}{50}$$
$$= 22.10 \text{ lbs. per I.H.P. hr.}$$

This, it should be noticed, is not the actual weight of steam used per horse power by the engine, but is that part which corresponds to the amount of dry steam remaining in the cylinder at the points under consideration. The amount is usually less when computed at cut-off than at the end of the stroke, since some of the steam which was condensed when the steam first entered the cylinder is restored by re-evaporation during the latter portion of the period of expansion.

349. The Diagram Water-rate for Multicylinder Engines. — The equations of the previous article apply only to simple engines. To compute the diagram water-rate for a compound or triple engine, proceed in exactly the same way for any given point on the expansion line of the cards from any of the cylinders, but after determining the steam accounted for per stroke, multiply by the number of strokes made per hour and *divide by the sum of the horse powers of all the cylinders.* This gives the steam credit for all the work it either has already done since entering the high-pressure cylinder or will do before leaving the low-pressure. This computation is equivalent to substituting for p in equation (5) what is known as the "equivalent mean effective pressure." Thus, if the computation is being made for the H.P. cylinder of a compound engine, the quantities above the line in equation (5) are chosen from the high-pressure card, while the

$$\text{Equiv. M.E.P.} = p' + rp''$$

where p' = M.E.P. of H.P. card,

p'' = M.E.P. of L.P. card, and

$$r = \text{ratio } \frac{\text{Vol. L.P. Piston-Displacement}}{\text{Vol. H.P. Piston-Displacement}}.$$

Similarly, if the computation is being made for the low-pressure card, the

$$\text{Equiv. M.E.P.} = \frac{p'}{r} + p''.$$

350. Approximate Formulas for the Diagram Water-rate. — If we put $l_a = l$, and further neglect all considerations of clearance, the final form of equation (5) would be

$$S = 13{,}750 \frac{w}{p}, \qquad (6)$$

in which p = the M.E.P. of the diagram, and w the weight of steam per cubic foot corresponding to terminal pressure. Several authorities have computed tables for this equation as follows:

Thompson's tables (see Appendix) give values of $13{,}750\ w$, and the tabular values must be divided by the M.E.P. to give the steam-consumption per I.H.P. per hour.

Tabor's tables give values of $\frac{13{,}750}{p}$, and the tabular values must be multiplied by the weight of a pound of steam corresponding to the terminal pressure, to give the steam-consumption.

Williams's tables, published in the Crosby catalogue, give values of $\frac{p}{425.43}$, and the results in each case have to be multiplied by $32.32\ w$ to give the steam-consumption.

A graphical correction may in all cases be made for the steam caught in clearance by drawing a horizontal line through the terminal pressure to compression line of diagram, and multiplying the result computed from the table by the ratio between the portion of this line intercepted between the terminal point and the compression line, to the whole stroke.

351. Re-evaporation and Cylinder Condensation. — By considering the hyperbolic curve as a standard, an idea can be obtained of the restoration by re-evaporation and the loss by cylinder condensation. Thus in Fig. 457, suppose that *a* is the point of cut-off at boiler-pressure. Construct an hyperbola as explained; in the example considered it is seen to lie above the expansion line for a short distance after cut-off, then to cross the line at *b*, and remain below it nearly to the end of the stroke. The amount by which the expansion line rises above the hyperbola may be considered as due to re-evaporation. The area of the diagram lying above would represent the work added by heat returned to the steam from the cylinder walls.

The methods for determining the cylinder condensation are similar to this process, except that the hyperbola is usually drawn upward from the point corresponding to the terminal pressure, to meet a horizontal line drawn to represent the boiler-pressure, as shown by the dotted lines in the diagrams in Fig. 460. The area of the figure enclosed by the dotted lines, compared with that of the diagram, is the ratio that the ideal diagram bears to the real; the difference is the loss by cylinder condensation.

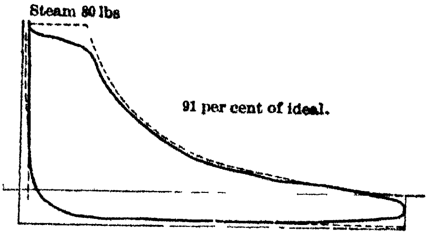


FIG 460

The student should understand that both these methods are approximations which may vary much from the truth.

352. Various Forms of Actual Indicator-diagrams. — The form of the indicator-card often reveals certain defects in steam distribution, due in most cases to improper valve setting. As a matter of fact, the main purpose of applying an indicator to an engine is in many cases merely to discover such defects in order to be able to eliminate them. A great variety of causes will affect the lines of the diagram and typical examples might be multiplied indefinitely. In many cases the cause of certain defects is easily located; in other cases, several causes may be at work and a certain amount of skill and experience may be necessary to interpret correctly.

If a card shows unusual features, the causes for them may be sought for in at least three directions: (*a*) faulty working of the indicator and faulty indicator connections, (*b*) errors in the setting of the valves, and (*c*) effects due to bad steam connections, leaky valves and pistons, etc.

(*a*) *Errors Due to Indicator and Faulty Indicator Connections.* — Some of these have already been discussed in the previous chapter under reducing motion, pencil motion, indicator pipe connections, etc. (See Chap. XV, Art. 332.)

Wavy lines in the expansion line, Fig. 461, may indicate spring vibration due to inertia effects or due to water in the piston and connections. The latter is the more likely cause, unless inertia effects also appear in other parts of the diagram.

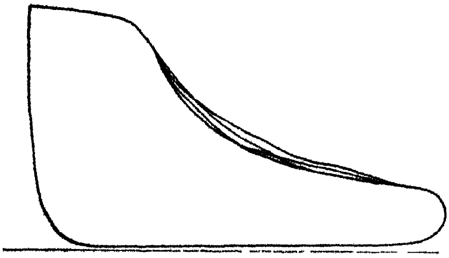


FIG. 461. — DIAGRAM SHOWING EFFECTS OF WATER IN INDICATOR CONNECTIONS.

The effect of faulty indicator connections usually manifests itself by throttling; see Fig. 462, where the broken line shows the normal diagram and the full line one which may result from too small or too long connections. The net result of this may be to make the diagram larger in area than it should be.

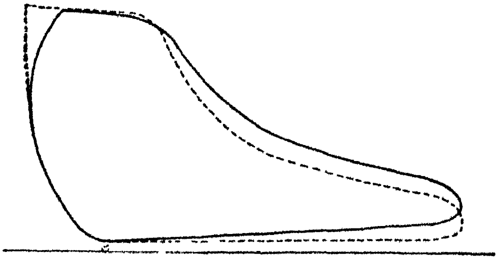
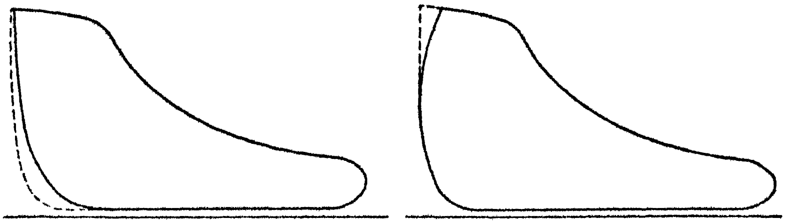


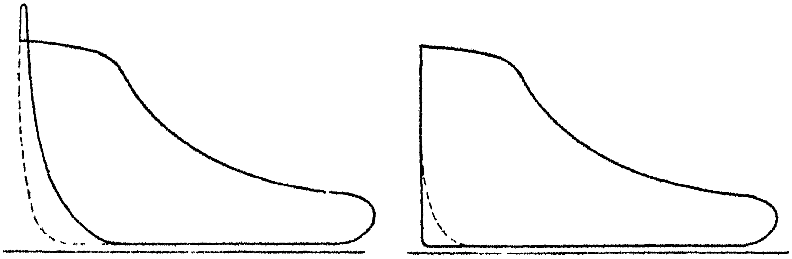
FIG. 462. — DIAGRAM RESULTING FROM FAULTY INDICATOR CONNECTIONS.

(*b*) *Errors Due to Valve Setting.* — Probably the great majority of unusual conditions in the lines of steam-engine cards is due to



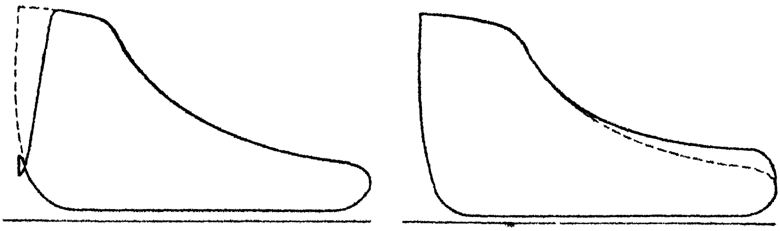
a Lead too Great, Admission too Early.

b Lead too Small, Admission too Late.



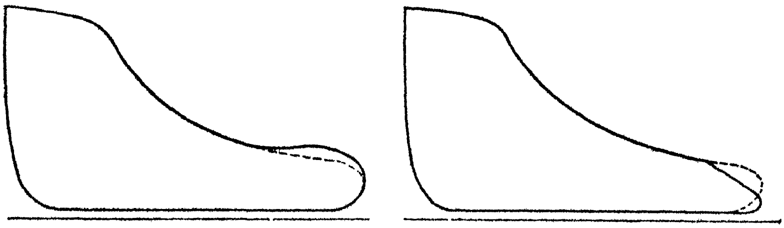
c Extreme Case of too High Compression

d Not Enough Compression.



e Late Admission of Steam.

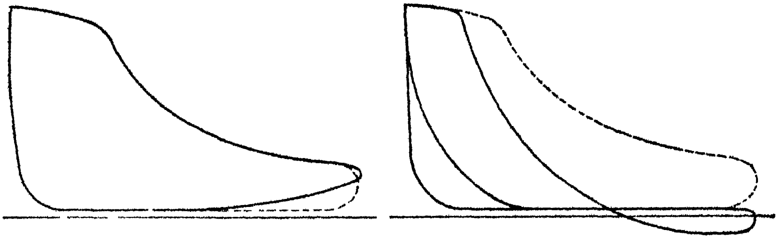
f Admission of Live Steam after Cut-off, Leaky Valves.



g Same as (*f*). May Occur in Valve Gears with Riding Cut-off.

h Too Early Release.

FIG. 463. (*a* to *h*.)

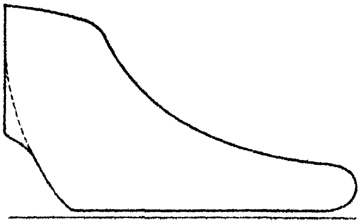


i

Too Late Release.

j

Loop Diagram due to Light Loads on Automatic Cut-off Engines.



k

Drop in Compression Line, most likely due to Leaky Pistons and Valves. Held in some cases to be due to condensation.

FIG. 463. (*i* to *k*.)

incorrect valve setting. There are many different valve gears, each likely to have its own peculiarities, but it will suffice to point out a few of the more usual abnormal conditions that may occur. In some of the following cases, Fig. 463, *a* to *k*, what may be considered the normal diagram is shown in dotted line, the actual diagram in full line.

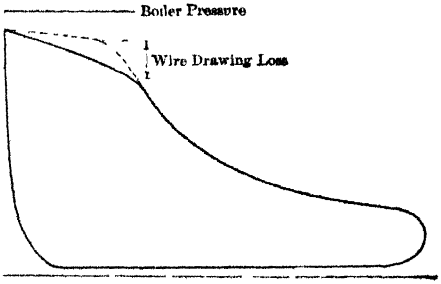


FIG. 464.

(*c*) *Effects Due to Bad Steam Connections, Leaky Valves and Pistons.* — Steam connections which are too small produce excessive wire drawing, (see Fig. 464).

A somewhat similar effect is produced in the use of valves which open and close too slowly.

Some of the effects of leaky pistons and valves have already been shown (Fig. 463, *f* and *k*). If a diagram shows an expansion line lying markedly below the hyperbola, Fig. 465, the effect may be due to serious leakage, steam leaking from the working to the exhaust side. If this loss is very serious it also appears in the compression line.

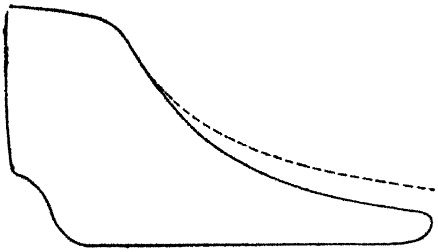


FIG. 465. — DIAGRAM SHOWING EFFECT OF SERIOUS PISTON AND VALVE LEAKAGE.

353. The Combining of Diagrams from Multicylinder Engines. — The methods used will be shown for the case of the compound engine, from which it is easy to generalize for triple or quadruple expansion engines.

There are two methods in use for combining indicator-cards. In the one the diagrams are each offset from the line of zero volumes in the combined diagram by a distance corresponding to the clearance volumes in the respective cylinders. In general, conditions will be such that the amount of steam expanding in the cylinder (which is in all cases equal to the sum of the weights of the steam taken into the engine per stroke plus the weight of steam caught in clearance) will not be the same in the two cylinders on account of the fact that the weight of clearance steam is not generally the same in the two cylinders. Hence it will not be possible to start with the H.P. cylinder and draw a single saturation curve for both cylinders, but such a curve must be drawn for each cylinder separately.

The second method of drawing the combined card is to eliminate the clearance steam and to set off from the line of zero volumes only the volumes of the steam taken into the engine per stroke. The method of doing this will be explained more fully later. In this construction a single saturation curve (or, for approximate work, an hyperbola) may be drawn for both cylinders, and some idea may thus be gained of how nearly the ideal work area represented under the saturation curve is realized by the actual area shown by the cards. But no quality computations can be made on such a diagram, and, for general usefulness, construction according to the method first outlined is generally preferred. Where the steam distribution in the two ends of the cylinder is fairly equal, it is usually satisfactory to average the two cards from the head and crank ends of the cylinder and to draw the combined diagrams

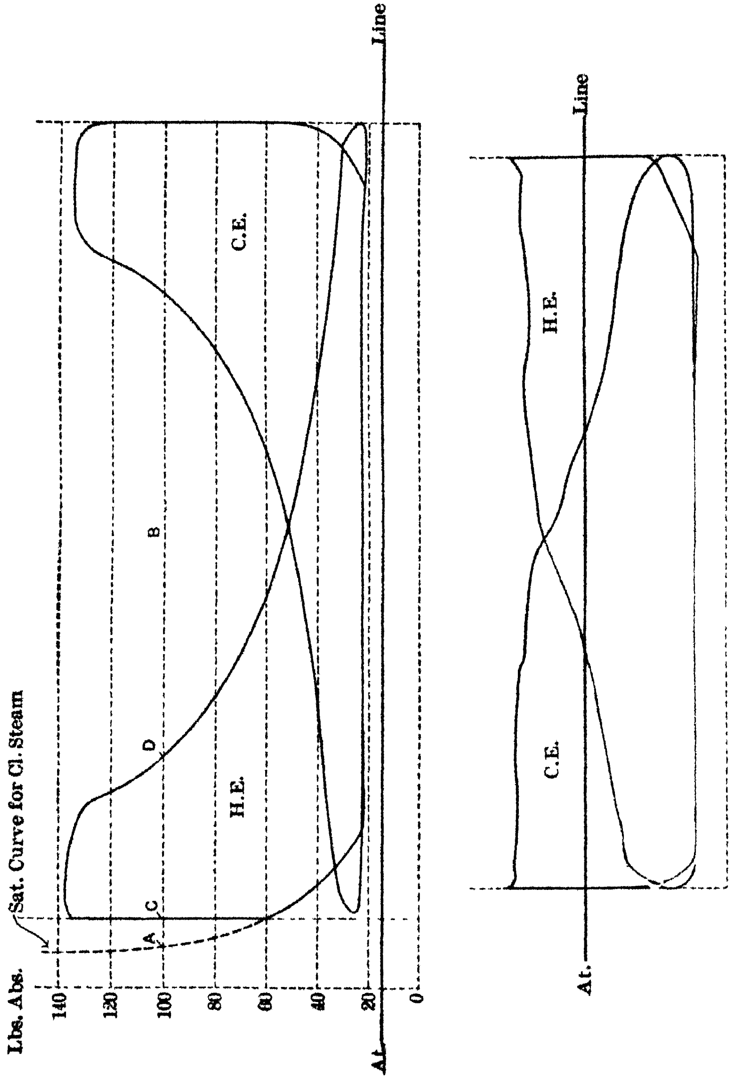


FIG. 466.—AVERAGE CARDS FROM A 9" BY 16" BY 36" CORLISS ENGINE.

from the average cards only. Where this cannot be done, it may be necessary to draw combined cards for the head and crank ends separately.

Figure 466 represents a set of cards taken from a cross-compound Corliss engine whose dimensions are as follows:

Dia. H.P. cyl	9 ins.
Dia. L.P. cyl	16 ins.
Length of stroke	36 ins.
Dia. piston rod, H.P. cyl	$2\frac{13}{16}$ ins.
Dia. piston rod, H.P. cyl	$2\frac{15}{16}$ ins.
Piston displacement, average: Head end and crank end, H.P. cyl	1.28 cu. ft.
Piston displacement, average: Head end and crank end, L.P. cyl	4.15 cu. ft.
Average clearance, H.P. cyl., per cent	7.59
Average clearance vol., H.P. cyl	.096 cu. ft.
Average clearance, L.P. cyl., per cent	8.93
Average clearance vol., L.P. cyl	.372 cu. ft.
Scale of indicator-spring, H.P. cyl	80 lbs.
Scale of indicator-spring, L.P. cyl	20 lbs.
Barometer, 28.75 in. Hg =	14.1 lbs.

CONSTRUCTION OF COMBINED CARD, METHOD I.—In Fig. 467 lay off on the X-axis a scale of volumes, and on the Y-axis a scale of absolute pressures as shown. Draw in the atmospheric line. Divide the length of each indicator-card, or the average card, into any convenient number of equal parts, say ten, and erect perpendiculars at the points of division, prolonging them downward until they reach the line of zero pressures in each case. In Fig. 467 lay off first the L.P. clearance volume (.372 cubic foot) and the volume of the L.P. cylinder (4.15 cubic feet), making a total of 4.522 cubic feet; divide the distance corresponding to 4.15 cubic feet also into ten equal parts, and erect ordinates. From the average L.P. card, Fig. 466, determine the pressure existing at the points of intersection of the ordinates for both forward and back pressure lines, measured above the line of zero pressure and using the proper spring scale. Lay in these pressures along the ordinates of Fig. 467, connect the points determined, and the result will be the L.P. diagram transferred to the new volume and pressure scales chosen.

Next, from the line of zero volume, Fig. 467, lay off the clearance volume of the H.P. cylinder (.096 cubic foot) and then the volume

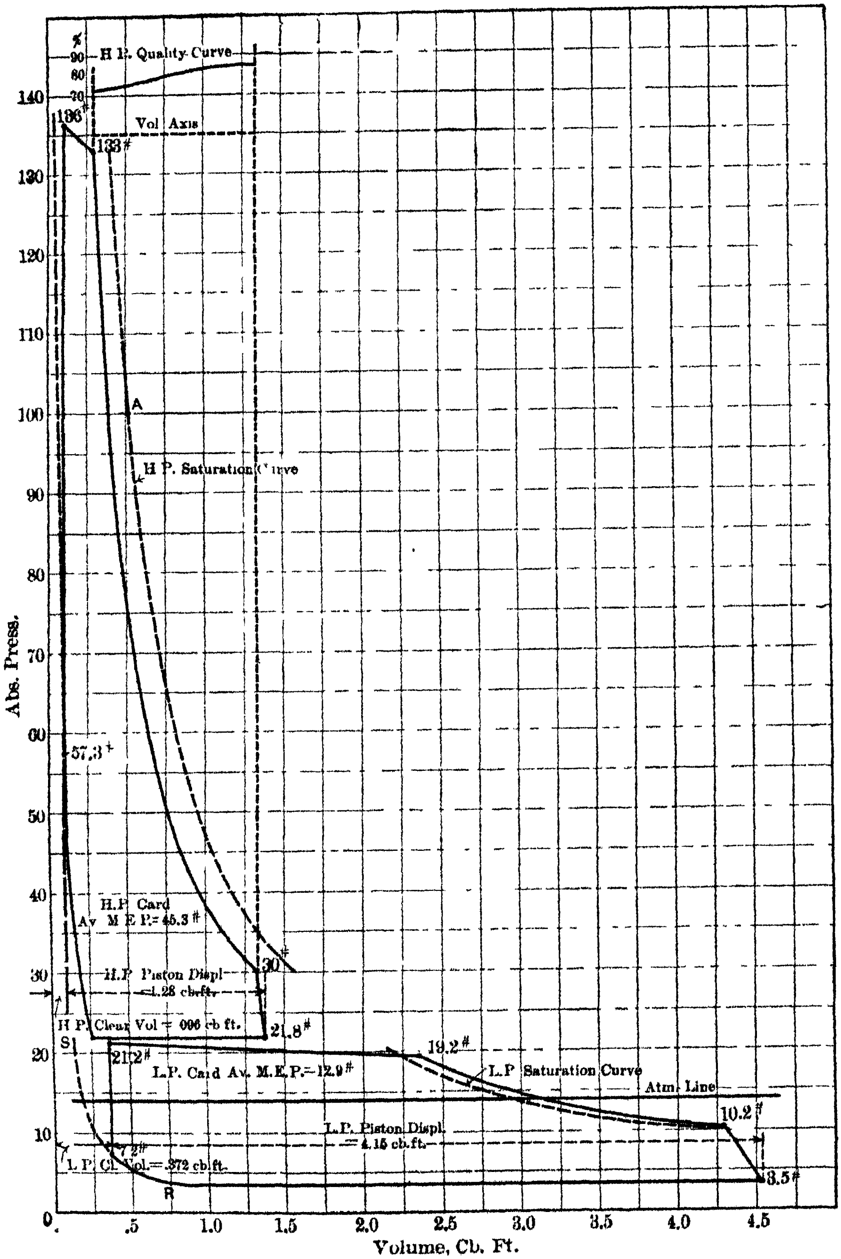


FIG. 467.—COMBINED DIAGRAM FOR COMPOUND STEAM ENGINE.

of the H.P. cylinder (1.28 cubic feet), making a total volume of 1.376 cubic feet; divide the distance corresponding to 1.28 cubic feet into ten equal parts and erect ordinates. Then proceed to determine the pressures from the average H.P. card, Fig. 466, measured above the line of zero pressure as a line of reference, and transfer these pressures to the ordinates of Fig. 467, as in the case of the L.P. card. Connecting the points will give the H.P. card.

If in any given case the lines of the original diagrams show more irregular variations than the sample cards chosen, it may be necessary to determine a greater number of pressure points by dividing the card lengths into a greater number of parts.

The Saturation Curves. — The next operation is to draw in the saturation curves. The steam taken into the engine per stroke for the test and the cards under consideration was .103 pound, as computed from water consumption and number of strokes.

The weight of steam caught in the clearance spaces may be computed from the pressure either at the beginning or end of the compression line, the corresponding volumes in the cylinder, and the specific volumes corresponding to the pressures, *on the assumption that the steam is dry and saturated.* There is some uncertainty regarding the quality of the steam along the compression line. Most authorities agree to call it dry and saturated at the beginning of compression. Hence the steam caught in clearance will in this case be found as follows for the H.P. cylinder:

Average absolute pressure at beginning of compression..	21.82 lbs.
Beginning of compression, per cent of back stroke......	88.3
Piston displacement during comp. = $(1.00 - .883)\ 1.28 =$	.15 cu. ft.
Clearance volume...........................	.096 cu. ft.
Total volume remaining at beginning of compression....	.246 cu. ft.
Sp. vol. per pound of steam at 21.82 lbs. (from steam-table)	18.52 cu. ft.
Weight of steam caught in clearance $= \frac{.246}{18.52} =$	.013 lb.

The total weight of steam in the H.P. cylinder at the beginning of expansion is therefore $= .103 + .013 = .116$ pound. The saturation curve can now be determined by finding the volume that this weight of steam, assuming it to remain dry, will occupy at different pressures, as shown in Article 346. For example, at 100 pounds absolute pressure, the specific volume of steam is 4.429 cubic feet; hence .116 pound of steam will occupy a volume of .514 cubic foot. This determines one point, marked A, on the saturation curve, Fig. 467.

The computations for the saturation curve for the L.P. cylinder are exactly similar. The engine steam per stroke is the same as before, = .103 pound. The weight of steam caught in the clearance is, however, in the L.P. cylinder only .006 pound as compared with .013 pound in the H.P. cylinder. The total weight of steam expanding is therefore = .109 pound. For this weight the volumes occupied at a series of different pressures are then found from the steam-table as before.

The Quality Curves. — The location of the saturation curve with reference to the expansion line shows the quality of the steam as the expansion progresses. In the case of the example in hand, the saturation curve for the H.P. cylinder lies entirely outside of the diagram and shows that the steam is wet throughout expansion. Since, at any given absolute pressure, the volume up to the expansion line shows the volume occupied by the steam in the mixture of steam and water contained in the cylinder, while the volume up to the saturation curve shows the volume that the weight of wet steam in the cylinder would have if it were just dry, it follows that the ratio of the total volume (including clearance) up to the expansion line to the total volume up to the saturation curve is a measure of the quality of the steam at the pressure chosen. Thus at the point of release in the H.P. cylinder, the pressure is 30.02 pounds and the volume is 1.325 cubic feet. At the same pressure, the saturation curve indicates a volume of 1.55 cubic feet. The quality at release is therefore $\frac{1.325}{1.55} = 85.5$ per cent. To get this result it is not at all necessary to determine these volumes; the ratio of the actual distances to the points on the two curves, measured say in hundredths of an inch, will give the same result. Making this computation for a number of points along the expansion line allows of the construction of a *quality* curve showing the variation of quality continuously. In this example this curve shows that the quality was 72 per cent at cut-off and increased gradually to 85 per cent at release, indicating a considerable amount of re-evaporation.

In the case of the L.P. diagram, the saturation curve lies entirely within the diagram, showing that the steam was superheated along the entire expansion line, due to the action of steam jackets which were not in use on the H.P. cylinder. The quality of the steam (i.e., the degree of superheat) can now no longer be determined by the simple ratio of volumes, because this relation no longer holds. The relation that may be used to find the temperature of the steam is the Tumlirz equation (Article 184):

$$v = 85.85\frac{T}{p} - .256$$

where v = specific volume of superheated steam in cubic feet,
T = absolute temperature,
p = absolute pressure in *pounds per square foot.*

To apply this to one point on the expansion line of the L.P. cylinder, take the volume at 17.5 pounds pressure. This equals 2.51 cubic feet for .109 pound of steam, so that the specific volume

$$v = \frac{2.51}{.109} = 23.03 \text{ cubic feet.}$$

Therefore

$$23.03 = 85.85\frac{T}{17.5 \times 144} - .256,$$

from which $T = 685^\circ$ F. abs. $= 225^\circ$ F. The steam-table shows that the critical temperature for 17.5 pounds is 220°. In this case the steam is therefore superheated only 5 degrees. This amount of superheat is practically negligible and is of course subject to the error of the measurements taken from the combined diagram and to the error of the Tumlirz equation which in itself is ± .8 per cent. It is therefore accurate enough to conclude in this case that the steam is dry and saturated along the expansion line, and therefore no quality curve for the L.P. cylinder need be drawn. In case superheat is shown to a greater extent, a quality curve with volumes as abscissæ and degrees of superheat as ordinates can be constructed similar to the quality curve for the H.P. cylinder.

CONSTRUCTION OF COMBINED CARD, METHOD II. — The first step in the process is to draw through the points of exhaust closure on each card a saturation curve for the weight of steam caught in the clearance. Then divide each diagram, that is an average for the H.P. cylinder and an average for the L.P. cylinder, into any convenient number of equally spaced horizontal strips, starting in each case from the line of zero pressures. Next on cross-section paper lay off scales of volumes and of pressures as for Method I. Lay in the atmospheric line and divide the field into as many horizontal strips as there are on the original diagrams, drawing the horizontal dividing lines at the proper pressure intervals by taking into account the scales of the springs.

In Fig. 466 the H.P. head end card shows the saturation curve drawn in for the clearance steam and also the method of horizontal subdivision. The next step is to determine the volumes along any horizontal line such as AB, Fig. 466, and to transfer them to the combined card. To do this by computation is rather cumbersome. Thus in the actual diagram, the length AC was .12 inch and the length AD was .92 inch. Since the actual full length of the card was 3.91 inches (which represented 1.28 cubic feet cylinder volume), the corresponding cylinder volumes are for AC .04 cubic foot and for AD .30 cubic foot. The line AB represents a pressure of 100 pounds absolute in this case. To transfer volumes AC and AD to the combined diagram, Fig. 468, lay off along the 100 pound pressure line the volumes .04 and .30 cubic foot, which determines the location of points C and D respectively, *the clearance steam being eliminated.*

To apply this method of computation to a number of pressure intervals, as would be necessary to obtain sufficient points to construct the diagram, is, as stated, very cumbersome, and for that reason a graphical method, given by Professor Heck, is preferable. In Fig. 469, lay off MN equal to the actual length of the H.P. diagram. Make NO equal to the actual distance on the combined diagram corresponding to 1.28 cubic feet (the volume of the H.P. cylinder). Draw OM. Then if we take any abcissa, as AD in the case of the H.P. head end card, Fig. 466, and transfer this to Fig. 469 (distance MN'), then will the distance $N'N''$ give the

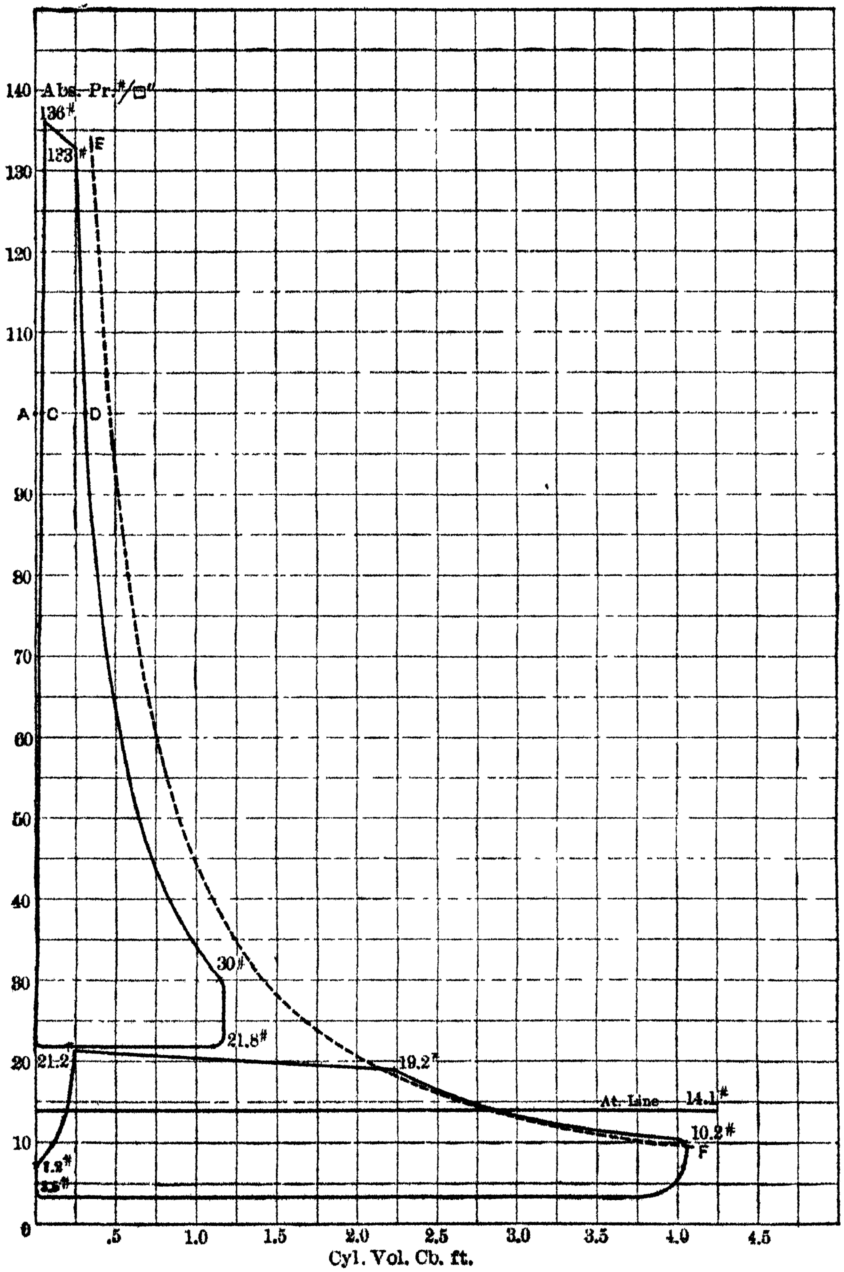


FIG. 468. — METHOD OF COMBINING INDICATOR CARDS, ELIMINATING CLEARANCE STEAM.

proper distance to be set off on the combined card for the volume *AD*. A similar construction figure may be drawn for the L.P. cylinder; in that case *MN* will represent the actual length of the L.P. card and *NO* the distance on the combined chart corresponding to 4.15 cubic feet, the volume of the L.P. cylinder.

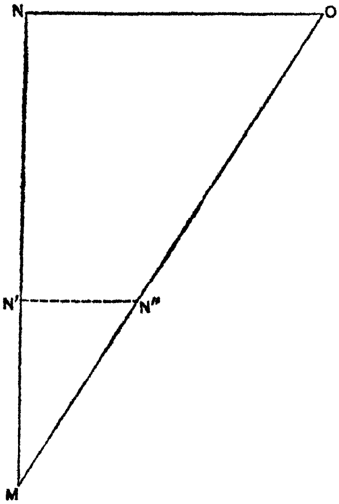


FIG. 469.

The final result of the construction of the combined card is shown in Fig. 468. To complete the diagram, draw in the saturation curve *EF*; in this case with a weight of steam equal to .103 pound, the engine steam.

In case the combined diagram according to Method I has already been drawn, the work of obtaining the diagram according to Method II is very much simplified, as it is merely necessary to draw in on the first combined card the saturation curves for the compression lines, starting with the point of exhaust closure (see curve *RS* in the L.P. card, Fig. 467), and then to set off from the line of zero volumes in Fig. 468 the volumes as measured from these saturation curves considered as lines of zero volume.

354. The Actual Entropy Diagram for a Steam Engine. — This diagram is sometimes used in the study of heat interchanges in the cylinder. It is, however, of doubtful utility for this purpose because we have definite information about the condition of the steam only along the expansion line. For every other line of the P–V diagram certain assumptions are ordinarily made which, depending upon conditions, may or may not be seriously wrong, and we obtain corresponding lines in the T–ϕ field subject to errors of the same uncertain magnitude. In effect, the usual method of transferring a diagram from the P–V to the T–ϕ field consists in determining the quality of the steam from a number of points around the P–V diagram, on the assumption:

(*a*) That the total weight of steam (engine + clearance) remains the same throughout the cycle, and (*b*) that the quality of this weight of steam for any point may be found by comparing the volume which it does actually occupy with that which it should occupy at the same pressure if it were dry and saturated

In other words, all condensing and evaporation processes are assumed to take place in the cylinder. Quality, pressure, or temperature are all that is required to determine entropy, and the transfer is thus very simple. But condition (*a*) only holds for the expansion line and is not true for all the other lines of the $P-V$ diagram, although the assumptions made give lines in the $T-\phi$ field which show the heat changes with fair approximation, except for the compression line, for which the real conditions are certainly far different from those assumed. Whenever assumption (*a*) is incorrect it of course affects assumption (*b*) in like manner.

The principle upon which the $T-\phi$ diagram in general rests has already been explained in Article 183. The method of transferring an actual card will be illustrated in connection with Fig. 470. This represents a diagram taken from a single-cylinder throttling engine. The data required are as follows:

Diameter of cylinder	6 ins.
Stroke	8 ins.
Clearance, average	12 per cent
Revolutions per minute	186
Scale of spring	40 lbs.
Pressure, absolute, steam pipe	129 lbs.
Pressure, absolute, steam chest	85 lbs.
Barometer	14.5 lbs.
Quality of steam, steam chest	98.8 per cent
Steam used per hour, from test	351 lbs.
Length of card	3.41 ins.
Derived quantities:	
Cylinder volume	.131 cu. ft.
Clearance volume	.016 cu. ft.
Total volume	.147 cu. ft.
Weight of engine steam per stroke	.0157 lb.
Weight of clearance steam per stroke	.0020 lb.
Total weight per stroke (engine + clearance)	.0177 lb.

The preliminary work consists in laying in the zero pressure line in Fig. 470 and dividing the card into horizontal strips at definite pressure intervals with zero pressure as the reference line. The intersections of the horizontal lines with the lines of the diagram give a number of definite points of reference. Where the changes of volume and pressure are rapid, as around the toe of the card,

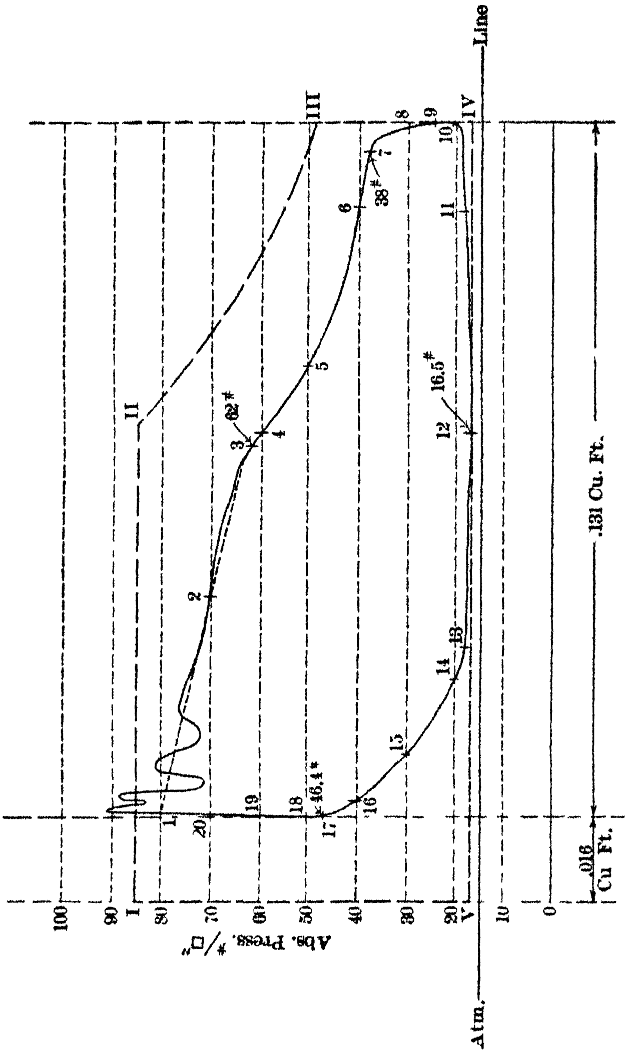


FIG. 470. — DIAGRAM FROM A SINGLE-CYLINDER THROTTLING ENGINE.

intermediate reference points may be chosen. In this particular case there are twenty such reference points (see Fig. 470). This card shows inertia waves in the admission line. In such cases it is common to draw in the average pressure line as shown.

Next construct a table (see below) with the columns headed as shown. Columns 1 and 2 are self-explanatory; the data for 3, 5, 8, and 10 are read directly from the steam-table. Column 4 is obtained by measurement from the indicator-card. For point 5, for example, the distance from the line of zero volumes is 2.64 inches. The total length of the card itself is 3.41 inches, which equals .131 cubic foot. Hence volume to point 5 $= \frac{2.64}{3.41} \times .131 = .1015$ cubic foot. Column 6 is obtained by dividing the volumes in column 4 by .0177, the total weight of steam, the result being the volume that this mixture of steam and water would occupy per pound. Column 7, the dryness fraction, is obtained by dividing column 6 by column 5. The method of finding column 9 is obvious, while column 11 is the sum of columns 9 and 10.

1	2	3	4	5	6	7	8	9	10	11
Reference Point.	Abs. Pressure. Lbs	Corr. Temp., t.	Volume to Point, Cu. Ft.	Specific Vol. per Lb Dry and Sat Steam.	Volume of Actual Steam per Lb.	Dryness Fraction, X.	Entropy of Evap., ϕ_r	$\phi_r \cdot X$.	Entropy of Liquid, ϕ_l	Total Entropy. $\phi_r \cdot X + \phi_l$
1	80	312	.0160	5 47	.91	.167	1.1665	.1912	4535	.644
2	70	303	.0575	6.20	3.25	.524	1.1896	6280	.4411	1.069
3	62	295	.0853	6 95	4 77	.688	1.2104	.8320	4302	1.262
4	60	293	.0877	7.17	4.95	.690	1.2160	8370	.4272	1.264
5	50	281	1015	8.51	5 73	.675	1.2468	.8410	.4113	1.252
6	40	267	.1323	10.49	7 48	.713	1 2841	9170	3920	1.309
7	38	264	.1420	11.01	8.05	.730	1.2950	.9470	.3877	1.335
8	30	250	.1460	13.74	8.24	.600	1 3311	.7960	.3680	1.164
9	25	240	.1470	16.30	8.31	.510	1 3604	.6930	.3532	1.046
10	20	228	.1470	20.08	8.31	.413	1.3965	.5780	.3355	.913
11	18	222	.1300	22.16	7 35	.331	1 4127	.4690	.3273	.796
12	16.5	218	.0890	24 00	5.03	.209	1.4261	.2980	.3208	.618
13	16.5	218	.0470	24.00	2.76	.115	1.4261	.1640	.3208	484
14	20	228	.0418	20 08	2.36	.117	1.3965	.1640	.3355	.499
15	30	250	.0271	13.74	1.53	.112	1.3311	.1480	.3680	.516
16	40	267	.0182	10 49	1.03	.098	1.2841	.1255	.3920	.517
17	46	276	.0160	9.18	.91	.099	1.2607	.1247	.4040	.531
18	50	281	.0160	8.51	.91	.107	1.2468	.1334	.4113	.544
19	60	293	.0160	7.17	.91	.127	1.2160	.1548	.4272	582
20	70	303	.0160	6.20	.91	.147	1.1896	.1745	.4411	.615

The last step consists in plotting the results in columns 3 and 11 (see Fig. 471). Choose any convenient scale for t and ϕ. The water line *AB* and the steam line *CD* are drawn directly by aid of steam-table data. Plotting the values in columns 3 and 11 gives the closed diagram shown, in which the points are marked with their proper reference number. This makes it easy to identify the lines with those on the *P–V* diagram.

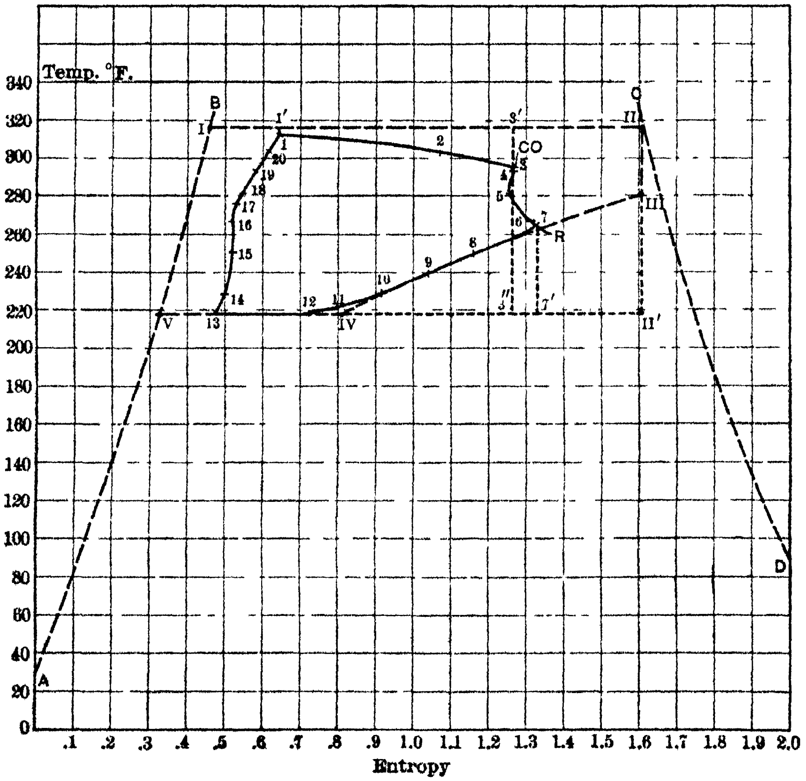


*FIG. 471 —ENTROPY DIAGRAM FOR CARD SHOWN IN FIG. 470.

To get some idea of the use of such a diagram, plot next what may be considered the theoretical (Rankine) diagram, assuming that the steam reaches cut-off under steam-chest conditions, that it then expands adiabatically to the end of the stroke, that next we have constant-volume exhaust to back pressure, and finally constant-pressure exhaust to the end of the stroke without compression. This diagram is marked I–II–III–IV–V on the *P–V* card, Fig. 470. The steam-chest pressure is 85 pounds, the quality 98.8 per cent, so that .0177 pound of this steam would locate the point II. These conditions in the steam chest also determine $t = 316$ degrees and total entropy $= 1.1561 \times .988 + .4590 = 1.602$, by means of which point II on the entropy diagram, Fig. 471, may be fixed. The rest of the theoretical diagram on the entropy chart is drawn by the method already explained for the real card.

In the real engine the steam in the steam chest, whose condition is defined by the point I, Fig. 471, due to wire drawing or passing through the valves and to initial condensation, changes its condition until at cut-off it is defined by point 3. The exact location of the line along which this change takes place is uncertain because of the lack of necessary data. It must be obvious from this and from the statements made above, that the exact itemizing of the heat losses and heat interchanges is hardly possible by means of such a diagram. Generally the interpretation is as follows:

The loss represented by the area 1-3-3′-1′, Fig. 471, is attributed to wire drawing.

The shape of the expansion line shows that heat is lost to the cylinder walls from 3 to 5, and that re-evaporation takes place from 5 to 7. The areas below 3-5 and 5-7 to the ϕ-axis (zero of absolute temperature) represent the amounts of the respective heat interchanges.

The area 3′-II-II′-3″ represents the loss due to initial condensation.

The loss due to early release is shown by the area 7-10-12-7′ for the real card and III-IV-II′ for the Rankine diagram.

The line 13-14 . . . . 20-1 shows a gain of entropy from the beginning, but no dependence should be placed in what this line seems to indicate on account of the arbitrary assumption made in obtaining it.

355. **Steam-chest and Steam-pipe Diagrams.** — These diagrams are sometimes taken in order to determine the pressure variations in steam chests and pipes. They are occasionally useful in bringing out wire drawing and other losses that may be due to insufficient pipe capacity. The method of taking them is exactly the same as for the cylinder diagram. Fig. 472 shows a steam-pipe and Fig. 473 a steam-chest diagram taken directly above the cylinder diagram

from a 30 inch × 60 inch Bass Corliss engine.* Concerning the last diagram, Barrus makes the point that since the wire-drawing loss in the cylinder card is accompanied by about the same loss of pressure in the chest, the trouble is not due to the valve but to some cause between the steam chest and the boiler. The steam-pipe diagram would indicate that the pipe is too small, although as a matter of fact the loss of pressure is not at all excessive.

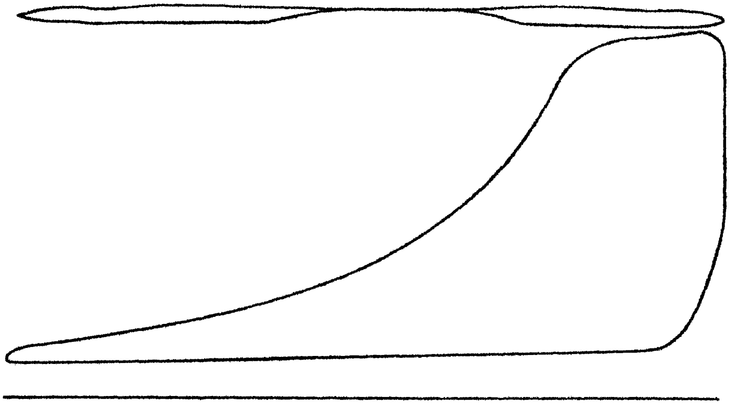


FIG. 472. — STEAM-PIPE DIAGRAM IN CONNECTION WITH INDICATOR CARD.

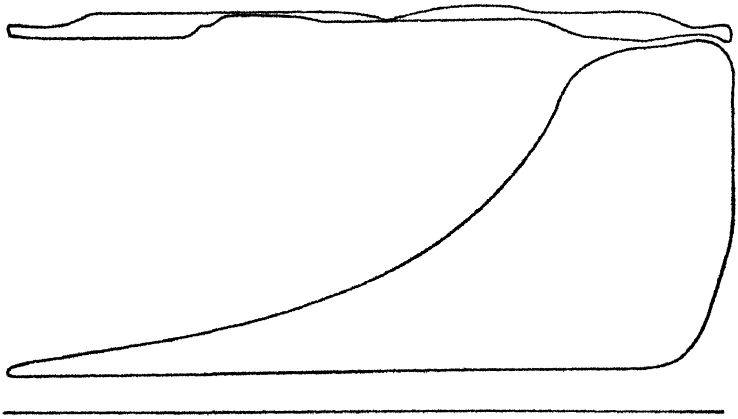


FIG. 473. — STEAM-CHEST DIAGRAM IN CONNECTION WITH INDICATOR CARD.

356. Displaced Diagrams and Time Diagrams. — The ordinary diagram is drawn with an harmonic motion, the speed being a maximum near the middle of the stroke, while near each end the speed is much less and is zero at the end of the stroke. This in a sense disguises the pressure variations occurring at the ends of the stroke. For certain purposes of study (explosion actions in gas engines, valve gear actions in steam engines, etc.) the pressure variations at the ends of the card are of the greatest importance, and in this connection displaced or distorted diagrams and time diagrams are of service.

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* Reproduced from Barrus, The Star Improved Indicator.

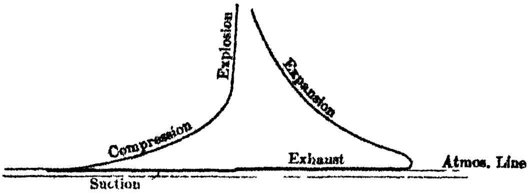


FIG. 474. — DISPLACED DIAGRAM FROM A GAS ENGINE.

In the case of displaced diagrams, the reducing motion is simply so constructed as to lag behind or lead the engine crank by a certain definite angle. If, for instance, the angle is made 90 degrees in the case of the gas engine, the diagram obtained will be somewhat of the shape of Fig. 474. The pressure variations during compression and explosions may from such a diagram be studied with much greater accuracy than from the regular type.

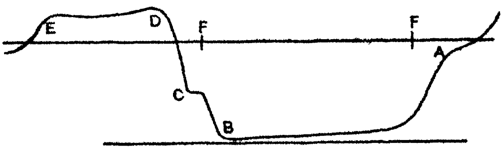


FIG. 475. — CRANK SHAFT DIAGRAM FROM A STEAM ENGINE.

The motion for an indicator may of course be taken from any moving part of the engine and occasionally crank-shaft diagrams, Fig. 475, are taken for the purpose of studying valve gear action. In such a case the distances moved by the drum are made proportional to the travel of the crank-pin, not to that of the piston. In the figure, line *AB* is the exhaust, *BC* compression, *CDE* admission, and *EA* expansion. Marks *FF* indicate the travel for one stroke (180 degrees).

In the time diagram, the oscillating motion of the drum is replaced by continuous uniform motion, by means of special apparatus (chronograph). For a gas engine a time diagram would look as indicated in Fig. 476.

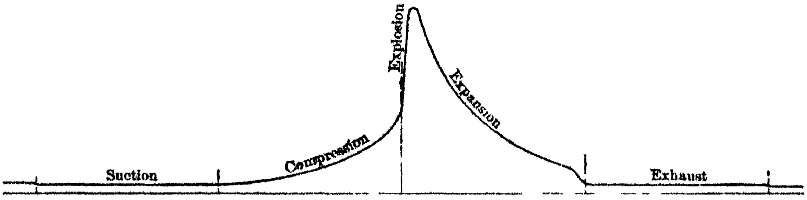


FIG. 476.—TIME DIAGRAM FROM A GAS ENGINE

It need hardly be mentioned that none of the displaced or time diagrams can be used for the determination of power without transposition.

II. GAS-ENGINE DIAGRAMS

357. Theoretical and Normal Diagrams. - The lines of the theoretical gas-engine diagrams and the efficiency formulas for these diagrams have already been discussed in Art. 185, Chap. XI. In that connection only the lines of the actual power diagram were considered In practice, however, normal diagrams show also suction and exhaust lines, at least in the case of 4-cycle engines, while in 2-cycle engines there must always be a "pump" diagram separate from the power cylinder card. The suction and exhaust lines of a 4-cycle diagram also enclose an area, known as the *lower-loop diagram*, and this diagram, or the pump diagram of the 2-cycle engine, represents the work required for scavenging and charging the power cylinder. The term " fluid " friction is sometimes used to designate the work so expended. It is of course not available for useful purposes, lowers the brake horse power developed, and is in that sense a loss.

In the normal 4-cycle gas-engine diagram, either from a constant-volume (Otto) engine or from a constant-pressure (Diesel) engine, the lower-loop diagram is not sufficiently clear to allow of integration for the purpose of determining the fluid friction loss, on account of the stiffness of indicator spring necessary to obtain the rest of the diagram. See Figs. 477 and 478. The former of these is from a 6-H.P. Hornsby-Akroyd oil engine, the latter from a 11½ inch × 18 inch Struthers-Wells natural-gas engine. The latter diagram shows the loop well. As might be expected, the Diesel engine card, Fig. 479, shows no indication of a lower loop on account of the extremely stiff spring that had to be used to stand the maximum pressure (over 500 pounds per square inch).

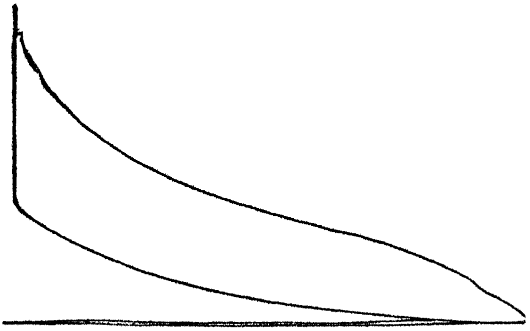


FIG. 477.—NORMAL DIAGRAM, 4-CYCLE OIL ENGINE.

To determine accurately the horse power lost in fluid friction, it becomes necessary to take lower-loop diagrams. This is done by using a weak spring and putting a "stop" on the piston-rod of the indicator piston, to prevent wrecking the spring. A good example of this kind of diagram, taken from a 7 inch × 9 inch Fairbanks gasoline engine, is shown in Fig. 480. The arrows show which way the lines were traced and the lines themselves are marked.

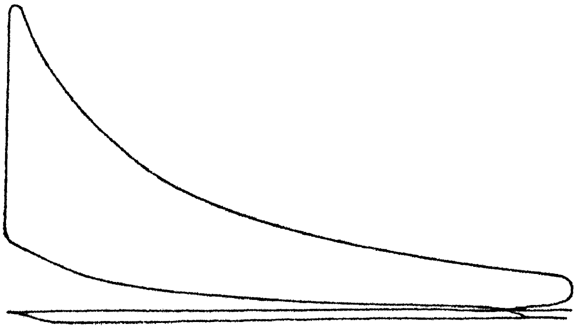


FIG. 478.—NORMAL DIAGRAM, 4-CYCLE GAS ENGINE.

Abnormal features may occur in gas-engine diagrams due to a variety of causes whose number is probably even greater than in the case of steam-engine cards. What has been said under steam-engine diagrams concerning the faults due to indicator, reducing motion and connections, applies with generally equal force also to gas-engine diagrams. Besides faulty valve timing, which is analogous to wrong valve setting in the steam engine, we have in the case of the gas engine also chances of error in wrong spark-timing, bad proportion of mixture, non-uniformity of mixture, etc. What can be accomplished by proper or improper control of mixture and spark is well shown in the series of cards, Fig. 481, *a* to *f*, which were taken from a 7 inch × 9 inch Fairbanks gasoline engine.

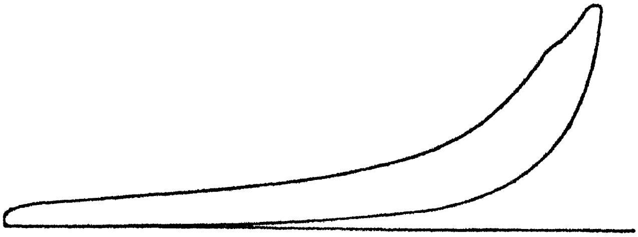


FIG. 479.—NORMAL DIAGRAM, DIESEL ENGINE.

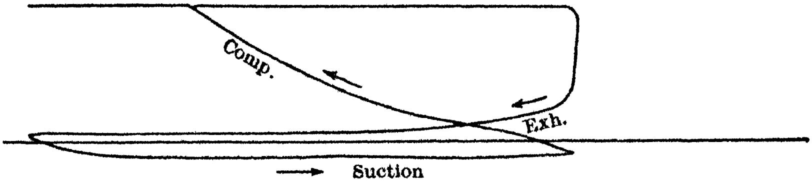
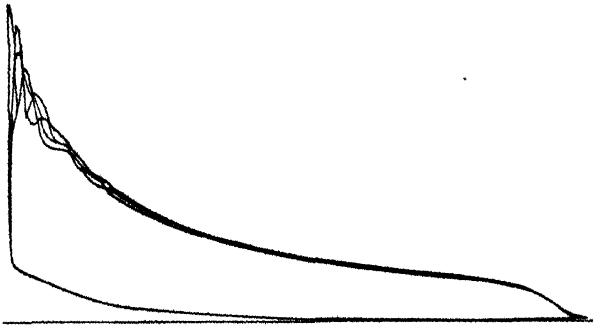


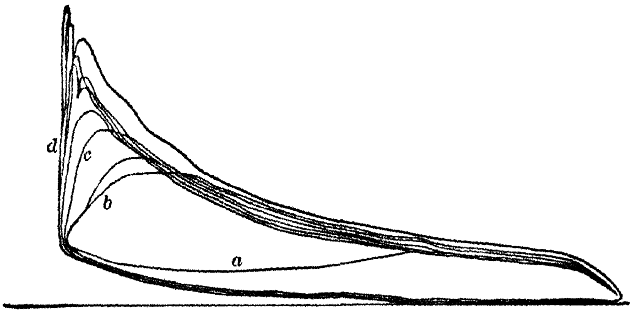
FIG. 480.—LOWER LOOP DIAGRAM, FAIRBANKS GASOLENE ENGINE.

Figs. 482, *a*, *b*, and *c* were taken from a 6-H.P. Hornsby-Akroyd oil engine. This engine ignites by compression, so that there is no spark control.

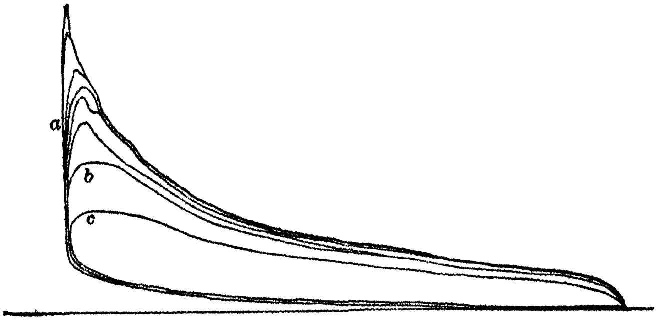
The most valuable aid to determine whether the valves are of proper size, or whether the valve timing is right, is the lower-loop diagram. Fig. 480 in this connection shows a normal lower loop. Figs. 483, *a*, *b*, and *c* were obtained from a Hornsby-Akroyd engine. Compare the magnitude of the loss in *b* and *c* with that in *a*. A



a.
Mixture normal. Spark normal.

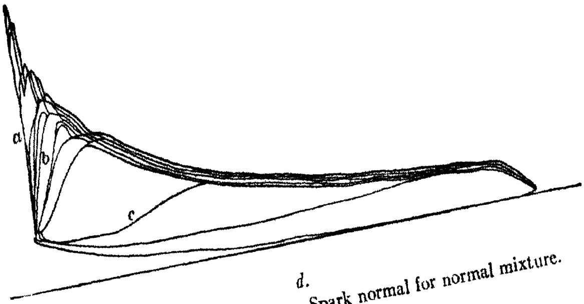


b.
Mixture lean. Spark changed from what would be normal for normal mixture to advanced position. Diagrams in order *a*, *b*, *c*, *d*.



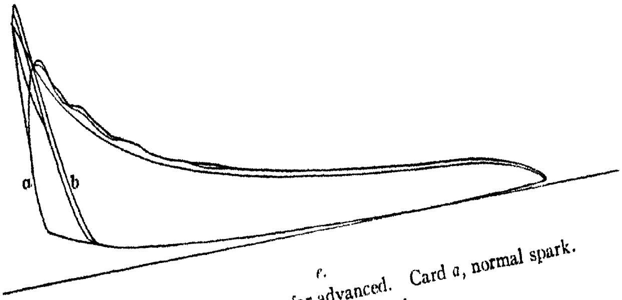
c.
Mixture changed normal to rich. Spark normal for normal mixture. Cards in order *a*, *b*, *c*.

FIG. 481. (*a* to *c*.)



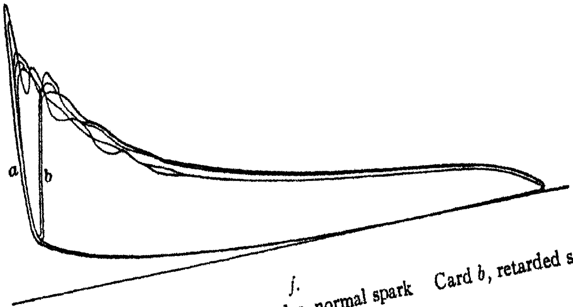
d.

Mixture changed normal to lean. Spark normal for normal mixture. Cards in order *a*, *b*, *c*.



e.

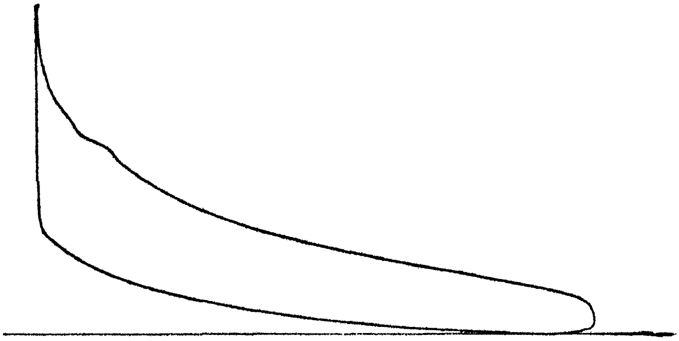
Mixture normal. Spark too far advanced. Card *a*, normal spark. Card *b*, advanced spark.



f.

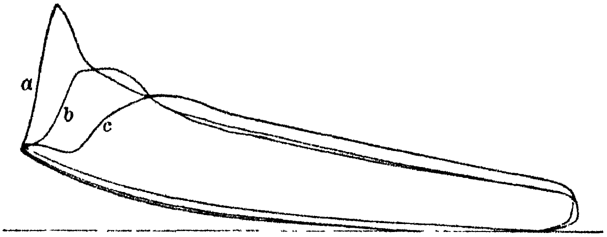
Mixture normal. Spark too late. Card *a*, normal spark Card *b*, retarded spark.

FIG. 481. (*d* to *f*.)



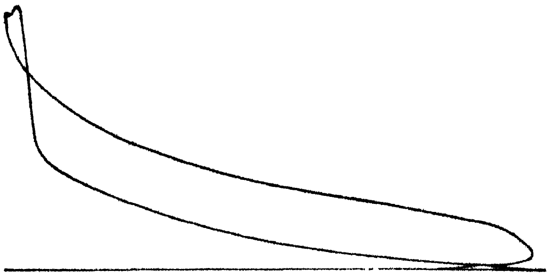
a.

Normal Card.



b.

Oil Supply normal. Air Supply gradually throttled. Cards in order *a*, *b*, *c*.

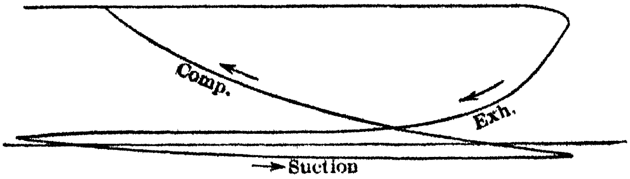


c.

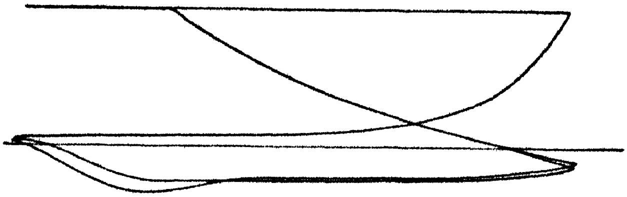
Case of Pre-ignition. Compression too high.

FIG. 482. (*a*, *b*, *c*.)

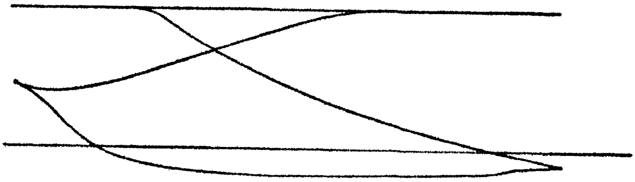
good example of the complexity of the actions that may take place during exhaust and suction with high speeds is shown in Fig. 484.



a.
Valve Setting about normal.



b.
Admission Valve opens too late.



c.
Exhaust Valve closes too early.
FIG. 483. (*a*, *b*, *c*.)

Inertia effects are shown on most of the cards in the series, Figs. 481, *a* to *f*. An extreme case of this is shown in Fig. 485. This

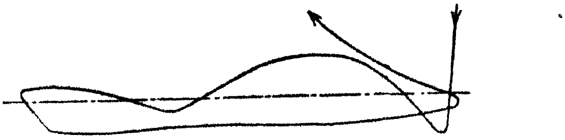


FIG. 484.

points directly to a bad choice of spring (generally too weak), although there is some controversy among authorities as to whether some indications of this kind are not due to a series of explosion waves correctly recorded by the indicator.

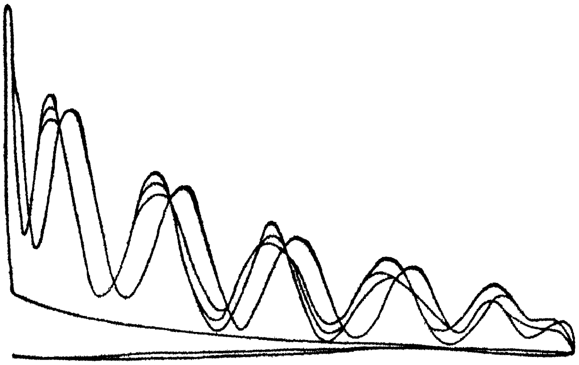


FIG. 485. — EXTREME CASE OF INERTIA EFFECT.

A typical 2-cycle diagram from the power cylinder is shown in Fig. 486. The exhaust port or valve opens at *c* and the pressure drops very quickly to nearly atmosphere. The new charge then enters, charging is complete by the time *b* is reached, when the exhaust and inlet valves close and compression begins. For further details see Chapter XXI on gas engines.

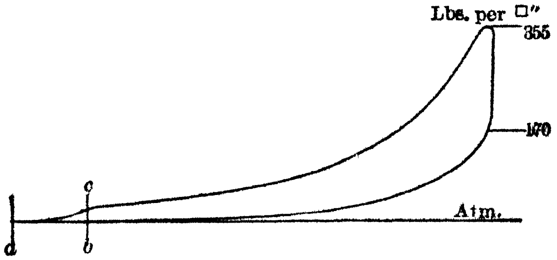


FIG. 486. — TYPE OF 2-CYCLE DIAGRAM.

358. The Actual Entropy Diagram for a Gas or a Mixture of Gases. — The entropy diagram for a gas engine is even more approximate than that for a steam engine, because in the present state of our knowledge we do not possess the means for a definite computation of heat quantities around the cycle. This is due to the fact that, although the laws of variation of the specific heats C_p and C_v with temperature are now fairly well known for the lower temperatures occurring in a gas engine cycle, considerable uncertainty still exists for temperatures over 2000° F. (See Chap. XXI.) The problem is further complicated by the complex actions along the combustion line, which, according to recent researches, make it probable that during this period the quantities of heat existing cannot be computed in the ordinary way by aid of values of C_p and C_v determined by experiments on mixtures of the same composition and corresponding temperatures, but unaccompanied by the energy (chemical) interchanges that occur along the combustion line. It is usual, in present practice, to avoid all these difficulties by computing C_p and C_v for the mixture as it enters the cylinder, and *then to assume that these remain constant throughout the cycle.* An improvement upon this method is to analyze the exhaust gases (or to compute the analysis approximately from the ratio of fuel to air used) and to compute C_p and C_v for the burned gases as well as for the fresh mixture from the specific heats of the component gases. Either method is an approximation only. Further approximations result from the fact that in the 4-cycle engine diagram the lower loop is generally neglected, and, in the case of the Diesel engine, the charge weight is not constant.

The simplest method* of transferring a $P-V$ diagram for a gas engine to the $T-\phi$ field is the following. To take a concrete case, Fig. 487 represents a card from a 7-H.P. 4-cycle illuminating gas engine.† This is the actual card transferred to cross-section paper because this transfer makes all measurements of pressure and volume direct. The lower loop (shaded area) is neglected. Draw a line g–a to close the diagram. From the engine test obtain the following data:

(*a*) Clearance and stroke volumes (.28 and .46 cubic foot).
(*b*) The ratio of air to fuel (13.8 by weight).

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* For a more elaborate method, see Carpenter & Diederichs Internal Combustion Engines, Chapter V.

† This card shows very low compression and explosion pressures, but will serve the purpose of showing the method of transfer as well as any other.

(*c*) The temperature at point *a*. If this is not known, it will have to be assumed. (739 degrees abs.).

(*d*) The values of C_p and C_v for the mixture at point *a*. These are assumed constant for the entire cycle ($C_p = .265$, $C_v = .191$).

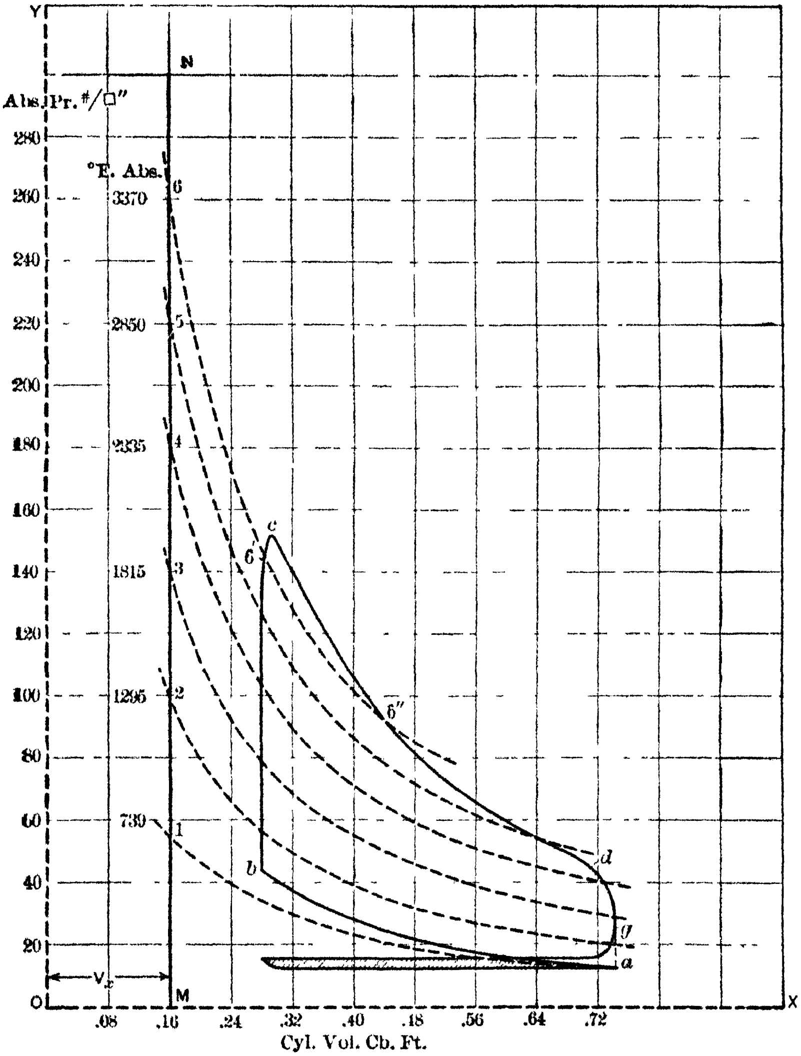

FIG 487.

Draw in the line of zero pressure *OX* and the line of zero volume *OY*. At any convenient volume division, V_x, draw a vertical. *MN*.

Through the point a, for which the temperature is known, draw an isothermal line (PV = const., with OX and OY as axes) to cut the line MN in point 1. Choose any other number of convenient pressure intervals along the line MN, as 2, 3, 4, etc., and through them draw isothermal lines. Each one of these cut the card in two points. Thus the isothermal drawn through the point 6 (260 pounds) cuts the card in 6′ and 6″.

Temperatures along the line MN can be computed (since temperature at point 1 is known) from the relation $\frac{P_2}{P_1} = \frac{T_2}{T_1}$; $\frac{P_3}{P_1} = \frac{T_3}{T_1}$, etc. (Note that all temperatures are absolute). This at once determines the temperatures for a number of points around the card, for $T_6 = T_{6'} = T_{6''}$.

To draw the entropy diagram for one pound of the mixture choose convenient scales of T and ϕ, see Fig. 488.

The general equation for a change of entropy in gases is

$$\phi_2 - \phi_1 = C_v \log_e \frac{T_2}{T_1} + (C_p - C_v) \log_e \frac{V_2}{V_1}.$$

The line MN is a constant-volume line, hence $\frac{V_2}{V_1} = 1$, and the second term of the right hand member in the above equation is 0. Also, if we assume arbitrarily that the entropy for 32 degrees is zero, the equation for any point, as 6, on the line MN becomes

$$\phi_6 = C_v \log_e \frac{T_6}{492}.$$

Compute a sufficient number of values for other points from this equation to locate the line MN, Fig. 488.

Next compute the entropy for a number of points on the diagram. To explain the method, take point 6′ as an example.

For the change from 6 to 6′, the first term of the general equation above becomes 0, since $T_6 = T_{6'}$, and the equation becomes

$$\phi_{6'} - \phi_6 = (C_p - C_v) \log_e \frac{V_{6'}}{V_x}.$$

Lay off the entropies thus computed from the line MN. Compute for sufficient points to locate the diagram, Fig. 488. The

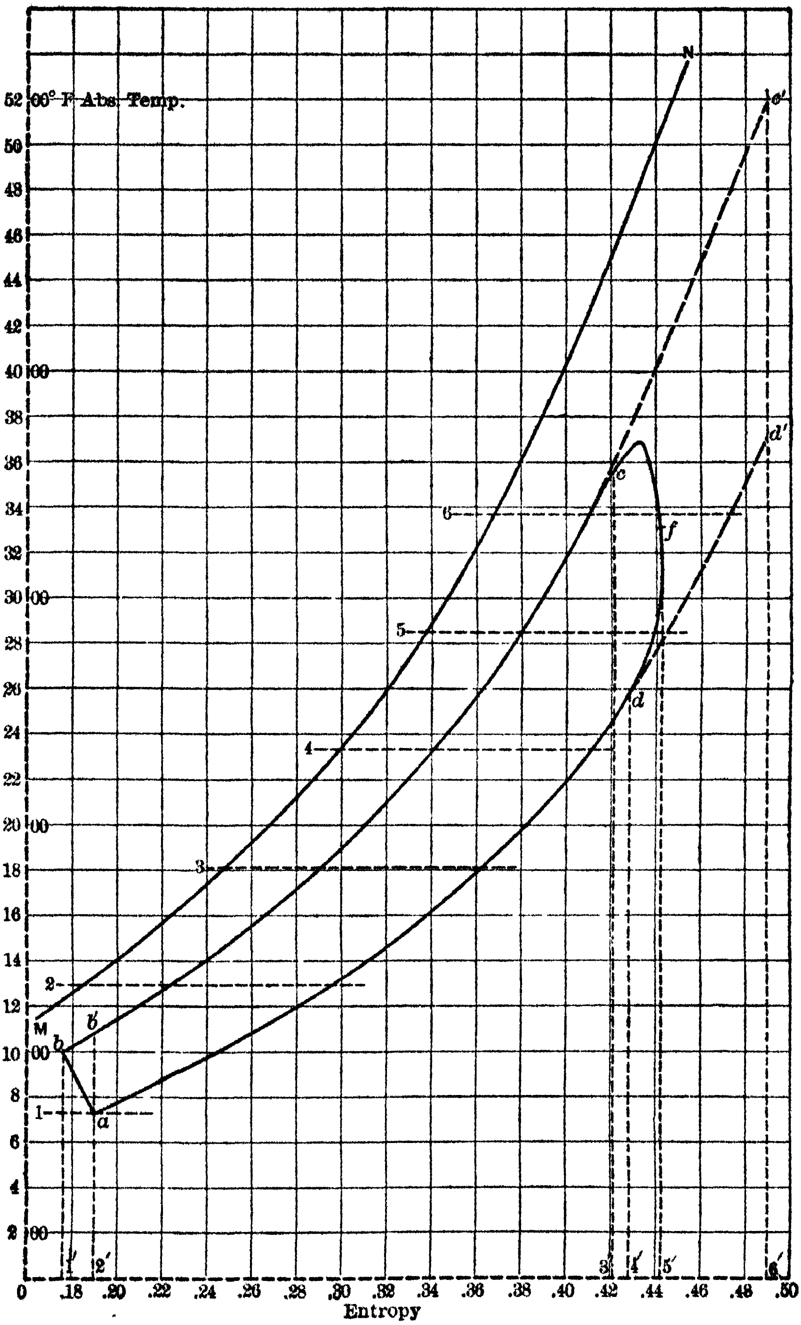

Fig. 488.—Entropy Diagram for Gas Engine.

resulting diagram is *abcd*, the points marked being those indicated by the same letters in Fig. 487.

To interpret this diagram it will be necessary to draw in the ideal Otto cycle diagram for the heat content of a pound of the mixture used, see Fig. 254, p. 349. This is indicated in Fig. 488 by the area *ab'c'd'a*. Then the various areas may be taken to represent the following heat quantities:

Area 1′ *b c* 3′ 1′ heat received along combustion line.
Area 3′ *c f* 5′ 3′ heat received during expansion.
Area 4′ *d f* 5′ 4′ heat lost to cylinder walls during expansion.
Area 2′ *a d* 4′ 2′ heat lost in exhaust.
Area 2′ *a b* 1′ 2′ heat lost in compression.
Area 4′ *d f c c′ d′* 6′ 4′ heat lost in radiation and conduction (jacket loss, etc.).

The last loss is very approximate only, since the assumption of constant specific heats affects this loss area very largely.

CHAPTER XVII.

THE TESTING OF STEAM BOILERS.

359. Methods of Testing Steam Boilers. — In 1884 the American Society of Mechanical Engineers adopted a code for the testing of steam boilers which was published in Vol. VI of the Transactions. This code was revised in 1899,* the full text, together with a number of appendices, being published in Vol. XXI. The Code is reproduced in full in the following pages. In the forty-one appendices attached to the Code in the Transactions the various members of the code committee express their views on certain provisions in greater detail, offer more detailed suggestions concerning certain operations, discuss instruments, apparatus, etc. To quote all of this material in full would occupy too much space, especially since many of the subjects mentioned have already been discussed in other parts of this book. But many of the remaining suggestions are valuable and to the point, and for that reason it is thought best to abstract these and to offer them as a sort of comment on the rules of the Code.

The Code has now been established some ten years. In that time there has been obtained more accurate information regarding some subjects, as, for instance, the heat content of steam. Further, the practice of many testing engineers differs somewhat, although in no important detail, from some of the rules. It was thought best, therefore, in commenting upon the rules, also to take into account these developments and varying viewpoints.

Whenever a rule has been thus annotated, the page reference to the note is given in every case. The notes themselves give chapter or page reference if the subject matter of the rule to which the note refers is treated in other parts of this book.

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* The Society has been at work on the revision of the codes for testing power plant apparatus, but the rules have not at this writing been finally approved. A preliminary revision was published in 1912.

RULES FOR CONDUCTING BOILER TRIALS.

CODE OF 1899.

I. *Determine at the outset* the specific object of the proposed trial, whether it be to ascertain the capacity of the boiler, its efficiency as a steam generator, its efficiency and its defects under usual working conditions, the economy of some particular kind of fuel, or the effect of changes of design, proportion, or operation; and prepare for the trial accordingly (see note, p. 710).

II. *Examine the boiler*, both outside and inside; ascertain the dimensions of grates, heating surfaces, and all important parts; and make a full record, describing the same, and illustrating special features by sketches. The area of heating surface is to be computed from the surfaces of shells, tubes, furnaces, and fire-boxes in contact with the fire or hot gases. The outside diameter of water-tubes and the inside diameter of fire-tubes are to be used in the computation. All surfaces below the mean water level which have water on one side and products of combustion on the other are to be considered as water-heating surface, and all surfaces above the mean water level which have steam on one side and products of combustion on the other are to be considered as superheating surface (see note, p. 710).

III. *Notice the general condition* of the boiler and its equipment, and record such facts in relation thereto as bear upon the objects in view.

If the object of the trial is to ascertain the maximum economy or capacity of the boiler as a steam generator, the boiler and all its appurtenances should be put in first-class condition. Clean the heating surface inside and outside, remove clinkers from the grates and from the sides of the furnace. Remove all dust, soot, and ashes from the chambers, smoke connections, and flues. Close air leaks in the masonry and poorly fitted cleaning doors. See that the damper will open wide and close tight. Test for air leaks by firing a few shovels of smoky fuel and immediately closing the damper, observing the escape of smoke through the crevices, or by passing the flame of a candle over cracks in the brickwork.

IV. *Determine the character of the coal* to be used. For tests of the efficiency or capacity of the boiler for comparison with other boilers the coal should, if possible, be of some kind which is commercially regarded as a standard. For New England and that portion of the country east of the Allegheny Mountains, good anthracite egg coal, containing not over 10 per cent of ash, and semi-bituminous Clearfield (Pa.), Cumberland (Md.), and Pocahontas (Va.) coals are thus regarded. West of the Allegheny Mountains, Pocahontas (Va.) and New River (W. Va.) semi-bituminous, and Youghiogheny or Pittsburg bituminous coals are recognized as standards.* There is no special grade of coal mined in the Western States which is widely recognized as of superior quality or considered as a standard coal for boiler testing. Big Muddy lump, and Illinois coal mined in Jackson County, Ill., is suggested as being of sufficiently high grade to answer these requirements in districts where it is more conveniently obtainable than the other coals mentioned above.

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* These coals are selected because they are about the only coals which possess the essentials of excellence of quality, adaptability to various kinds of furnaces, grates, boilers, and methods of firing, and wide distribution and general accessibility in the markets. See various appendices in Vol. XXI, Transactions A. S. M. E.

For tests made to determine the performance of a boiler with a particular kind of coal, such as may be specified in a contract for the sale of a boiler, the coal used should not be higher in ash and in moisture than that specified, since increase in ash and moisture above a stated amount is apt to cause a falling off of both capacity and economy in greater proportion than the proportion of such increase (see note, p. 711).

V. *Establish the correctness of all apparatus* used in the test for weighing and measuring. These are:

1. Scales for weighing coal, ashes, and water.
2. Tanks, or water meters, for measuring water. Water meters, as a rule, should only be used as a check on other measurements. For accurate work, the water should be weighed or measured in a tank. (See Chapter XII.)
3. Thermometers and pyrometers for taking temperatures of air, steam, feed-water, waste gases, etc. (Chapter VII.)
4. Pressure-gauges, draught-gauges, etc. (Chapter VI.)

The kind and location of the various pieces of testing apparatus must be left to the judgment of the person conducting the test; always keeping in mind the main object, i.e., to obtain authentic data. (See note, p. 711.)

VI. *See that the boiler is thoroughly heated* before the trial to its usual working temperature. If the boiler is new and of a form provided with a brick setting, it should be in regular use at least a week before the trial, so as to dry and heat the walls. If it has been laid off and has become cold, it should be worked before the trial until the walls are well heated.

VII. *The boiler and connections* should be proved to be free from leaks before beginning a test, and all water connections, including blow and extra feed pipes, should be disconnected, stopped with blank flanges, or bled through special openings beyond the valves, except the particular pipe through which water is to be fed to the boiler during the trial. During the test the blow-off and feed pipes should remain exposed to view. (See note, p. 711)

If an injector is used, it should receive steam directly through a felted pipe from the boiler being tested.*

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* In feeding a boiler undergoing test with an injector taking steam from another boiler, or from the main steam pipe from several boilers, the evaporative results may be modified by a difference in the quality of the steam from such source compared with that supplied by the boiler being tested, and in some cases the connection to the injector may act as a drip for the main steam pipe. If it is known that the steam from the main pipe is of the same pressure and quality as that furnished by the boiler undergoing the test, the steam may be taken from such main pipe.

If the water is metered after it passes the injector, its temperature should be taken at the point where it leaves the injector. If the quantity is determined before it goes to the injector the temperature should be determined on the suction side of the injector, and if no change of temperature occurs, other than that due to the injector, the temperature thus determined is properly that of the feed-water. When the temperature changes between the injector and the boiler, as by the use of a heater or by radiation, the temperature at which the water enters and leaves the injector and that at which it enters the boiler should all be taken. In that case the weight to be used is that of the water leaving the injector, computed from the heat units if not directly measured, and the temperature, that of the water entering the boiler.

Let w = weight of water entering the injector.
x = " " steam " " "
h_1 = heat units per pound of water entering injector.
h_2 = " " " " " steam " "
h_3 = " " " " " water leaving "
Then, $w + x$ = weight of water leaving injector.

$$x = w\frac{h_3 - h_1}{h_2 - h_3}.$$

See that the steam main is so arranged that water of condensation cannot run back into the boiler.

VIII. *Duration of the Test.* — For tests made to ascertain either the maximum economy or the maximum capacity of a boiler, irrespective of the particular class of service for which it is regularly used, the duration should be at least 10 hours of continuous running. If the rate of combustion exceeds 25 pounds of coal per square foot of grate surface per hour, it may be stopped when a total of 250 pounds of coal has been burned per square foot of grate.

In cases where the service requires continuous running for the whole 24 hours of the day, with shifts of firemen a number of times during that period, it is well to continue the test for at least 24 hours.

When it is desired to ascertain the performance under the working conditions of practical running, whether the boiler be regularly in use 24 hours a day or only a certain number of hours out of each 24, the fires being banked the balance of the time, the duration should not be less than 24 hours.

IX. *Starting and Stopping a Test.* — The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same; the water level the same; the fire upon the grates should be the same in quantity and condition; and the walls, flues, etc., should be of the same temperature. Two methods of obtaining the desired equality of conditions of the fire may be used, viz.: those which were called in the Code of 1885 " the standard method " and " the alternate method," the latter being employed where it is inconvenient to make use of the standard method.* (See note, p. 713.)

X. *Standard Method of Starting and Stopping a Test.* — Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time and the water level† while the water is in a quiescent state, just before lighting the fire.

At the end of the test remove the whole fire, which has been burned low, clean the grates and ash pit, and note the water level when the water is in a quiescent state and record the time of hauling the fire. The water level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and not by operating the pump after the test is completed.

XI. *Alternate Method of Starting and Stopping a Test.* — The boiler being thoroughly heated by a preliminary run, the fires are to be burned low and well cleaned. Note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the water level Note the time, and record it as the starting time. Fresh coal which has been weighed should now be fired. The ash pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave a bed of coal on the grates of the same depth, and in the same condition, as at the start. When this stage is reached, note the time and record it as the stopping time. The water level and steam pressures should previously be brought as nearly as possible to the same point as at the start. If the water level is not the same as at the start, a correction should be made by computation, and not by operating the pump after the test is completed.

XII. *Uniformity of Conditions.* — In all trials made to ascertain maximum economy or capacity, the conditions should be maintained uniformly constant. Arrangements should be made to dispose of the steam so that the rate of evaporation may be kept the same from beginning to end. This may be accomplished in a single boiler by carrying the steam through a waste steam pipe, the discharge from which can be regulated as desired. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers, leaving the test boiler to work under a constant rate of production.

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* The Committee concludes that it is best to retain the designations "standard" and "alternate," since they have become widely known and established in the minds of engineers and in the reprints of the Code of 1885. Many engineers prefer the "alternate" to the "standard" method on account of it being less liable to error due to cooling of the boiler at the beginning and end of a test.

† The gauge-glass should not be blown out within an hour before the water level is taken at the beginning and end of a test, otherwise an error in the reading of the water level may be caused by a change in the temperature and density of the water in the pipe leading from the bottom of the glass into the boiler.

Uniformity of conditions should prevail as to the pressure of steam, the height of water, the rate of evaporation, the thickness of fire, the times of firing and quantity of coal fired at one time, and as to the intervals between the times of cleaning the fires.

The method of firing to be carried on in such tests should be dictated by the expert or person in responsible charge of the test, and the method adopted should be adhered to by the fireman throughout the test.

XIII. *Keeping the Records.* — Take note of every event connected with the progress of the trial, however unimportant it may appear. Record the time of every occurrence and the time of taking every weight and every observation.

The coal should be weighed and delivered to the fireman in equal proportions, each sufficient for not more than one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the last of each portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler, and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the test may be divided into several periods if desired, and the degree of uniformity of combustion, evaporation, and economy analyzed for each period. In addition to these records of the coal and the feed-water, half-hourly observations should be made of the temperature of the feed-water, of the flue gases, of the external air in the boiler-room, of the temperature of the furnace when a furnace pyrometer is used, also of the pressure of steam, and of the readings of the instruments for determining the moisture in the steam. A log should be kept on properly prepared blanks containing columns for record of the various observations. (See note, p. 714, also p. 716.)

When the "standard method" of starting and stopping the test is used, the hourly rate of combustion and of evaporation and the horse power should be computed from the records taken during the time when the fires are in active condition. This time is somewhat less than the actual time which elapses between the beginning and end of the run. The loss of time due to kindling the fire at the beginning and burning it out at the end makes this course necessary.

XIV. *Quality of Steam.* — The percentage of moisture in the steam should be determined by the use of either a throttling or a separating steam calorimeter. The sampling nozzle should be placed in the vertical steam pipe rising from the boiler. It should be made of ½-inch pipe, and should extend across the diameter of the steam pipe to within half an inch of the opposite side, being closed at the end and perforated with not less than twenty ⅛-inch holes equally distributed along and around its cylindrical surface, but none of these holes should be nearer than ½ inch to the inner side of the steam pipe. The calorimeter and the pipe leading to it should be well covered with felting. Whenever the indications of the throttling or separating calorimeter show that the percentage of moisture is irregular, or occasionally in excess of three per cent, the results should be checked by a steam separator placed in the steam pipe as close to the boiler as convenient, with a calorimeter in the steam pipe just beyond the outlet from the separator. The drip from the separator should be caught and weighed, and the percentage of moisture computed therefrom added to that shown by the calorimeter.

Superheating should be determined by means of a thermometer placed in a mercury well inserted in the steam pipe. The degree of superheating should be taken as the difference between the reading of the thermometer for superheated steam and the readings of the same thermometer for saturated steam at the same pressure as determined by a special experiment, and not by reference to steam tables. (See note, p. 717.)

XV. *Sampling the Coal and Determining its Moisture.* — As each barrow load or fresh portion of coal is taken from the coal pile, a representative shovelful is selected from it and placed in a barrel or box in a cool place and kept until the end of the trial. The samples are then mixed and broken into pieces not exceeding one inch in diameter, and reduced by the process of repeated quartering and crushing until a final sample weighing about five pounds is obtained, and the size of the larger pieces is such that they will pass through a sieve with ¼-inch meshes. From this sample two one-quart, air-tight glass preserving jars, or other air-tight vessels which will prevent the escape of moisture from the sample, are to be promptly filled, and these samples are to be kept for subsequent determinations of moisture and of heating value and for chemical analyses. During the process of quartering, when the sample has been reduced to about 100 pounds, a quarter to a half of it may be taken for an approximate determination of moisture. This may be made by placing it in a shallow iron pan, not over three inches deep, carefully weighing it, and setting the pan in the hottest place that can be found on the brickwork of the boiler setting or flues, keeping it there for at least 12 hours, and then weighing it. The determination of moisture thus made is believed to be approximately accurate for anthracite and semi-bituminous coals, and also for Pittsburg or Youghiogheny coal; but it cannot be relied upon for coals mined west of Pittsburg, or for other coals containing inherent moisture. For these latter coals it is important that a more accurate method be adopted. The method recommended by the Committee for all accurate tests, whatever the character of the coal, is described as follows:

Take one of the samples contained in the glass jars, and subject it to a thorough air-drying, by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill adjusted so as to produce somewhat coarse grains (less than $\frac{1}{16}$-inch), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams, weigh it in a balance which will easily show a variation as small as 1 part in 1000, and dry it in an air or sand bath at a temperature between 240 and 280 degrees Fahr. for one hour. Weigh it and record the loss, then heat and weigh it again repeatedly, at intervals of an hour or less, until the minimum weight has been reached and the weight begins to increase by oxidation of a portion of the coal. The difference between the original and the minimum weight is taken as the moisture in the air-dried coal. This moisture test should preferably be made on duplicate samples, and the results should agree within 0.3 to 0.4 of one per cent, the mean of the two determinations being taken as the correct result. The sum of the percentage of moisture thus found and the percentage of surface moisture previously determined is the total moisture. (See note, p. 717.)

XVI. *Treatment of Ashes and Refuse.* — The ashes and refuse are to be weighed in a dry state. If it is found desirable to show the principal characteristics of the ash, a sample should be subjected to a proximate analysis and the actual amount of incombustible material determined. For elaborate trials a complete analysis of the ash and refuse should be made. (See note, p. 718)

XVII. *Calorific Tests and Analysis of Coal.* -- The quality of the fuel should be determined either by heat test or by analysis, or by both.

The rational method of determining the total heat of combustion is to burn the sample of coal in an atmosphere of oxygen gas, the coal to be sampled as directed in Article XV of this code. (See Chapter XIII.)

The chemical analysis of the coal should be made only by an expert chemist. The total heat of combustion computed from the results of the ultimate analysis may be obtained by the use of Dulong's formula (with constants modified by recent determinations), viz.: $14,540\ C + 61,950 \left(H - \frac{O}{8}\right) + 4000\ S$, in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur respectively, as determined by the ultimate analysis.* (See note, p. 719.)

It is desirable that a proximate analysis should be made, thereby determining the relative proportions of volatile matter and fixed carbon. These proportions furnish an indication of the leading characteristics of the fuel, and serve to fix the class to which it belongs. As an additional indication of the characteristics of the fuel, the specific gravity should be determined.

XVIII. *Analysis of Flue Gases.* — The analysis of the flue gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analyses great care should be taken to procure average samples — since the composition is apt to vary at different points of the flue. The composition is also apt to vary from minute to minute, and for this reason the drawings of gas should last a considerable period of time. Where complete determinations are desired, the analyses should be intrusted to an expert chemist. For approximate determinations the Orsat* or the Hempel† apparatus may be used by the engineer. (See Chapter XIII for treatment of methods, computations, etc.)

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* Favre and Silberman give 14,544 B.t.u. per pound carbon; Berthelot, 14,647 B.t.u. Favre and Silberman give 62,032 B.t.u. per pound hydrogen; Thomson, 61,816 B.t.u.

For the continuous indication of the amount of carbonic acid present in the flue gases, an instrument may be employed which shows the weight of the sample of gas passing through it.

XIX. *Smoke Observations.* — It is desirable to have a uniform system of determining and recording the quantity of smoke produced where bituminous coal is used. The system commonly employed is to express the degree of smokiness by means of percentages dependent upon the judgment of the observer. The Committee does not place much value upon a percentage method, because it depends so largely upon the personal element, but if this method is used, it is desirable that, so far as possible, a definition be given in explicit terms as to the basis and method employed in arriving at the percentage. The actual measurement of a sample of soot and smoke by some form of meter is to be preferred. (See Chapter XIII, under smoke determination.)

XX. *Miscellaneous.* — In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are in general unnecessary for ordinary tests. These are the measurement of the air supply, the determination of its contained moisture, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water. (See note, p. 719.)

As these determinations are rarely undertaken, it is not deemed advisable to give directions for making them.

XXI *Calculations of Efficiency.* — Two methods of defining and calculating the efficiency of a boiler are recommended. They are:

1. $$\text{Efficiency of the boiler} = \frac{\text{Heat absorbed per lb. combustible}}{\text{Calorific value of 1 lb. combustible}}.$$

2. $$\text{Efficiency of the boiler and grate} = \frac{\text{Heat absorbed per lb. coal}}{\text{Calorific value of 1 lb. coal}}.$$

The first of these is sometimes called the efficiency based on combustible, and the second the efficiency based on coal. The first is recommended as a standard of comparison for all tests, and this is the one which is understood to be referred to when the word "efficiency" alone is used without qualification. The second, however, should be included in a report of a test, together with the first, whenever the object of the test is to determine the efficiency of the boiler and furnace together with the grate (or mechanical stoker), or to compare different furnaces, grates, fuels, or methods of firing.

The heat absorbed per pound of combustible (or per pound of coal) is to be calculated by multiplying the equivalent evaporation from and at 212 degrees per pound combustible (or coal) by 965.7. (See note, p. 719, concerning **this rule, the following rule, and the Tables below.**)

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* See R. S. Hale's paper on "Flue Gas Analysis," Transactions, Vol. XVIII, p. 901.

† See Hempel's "Methods of Gas Analysis," translated by L. M. Dennis (Macmillan & Co.).

XXII. *The Heat Balance.* — An approximate "heat balance," or statement of the distribution of the heating value of the coal among the several items of heat utilized and heat lost, may be included in the report of a test when analyses of the fuel and of the chimney gases have been made. It should be reported in the following form:

HEAT BALANCE, OR DISTRIBUTION OF THE HEATING VALUE OF THE COMBUSTIBLE.

Total Heat value of 1 lb. of combustible.... .... B.t.u.

	B.t.u.	Per Cent.
1. Heat absorbed by the boiler = evaporation from and at 212 degrees per pound of combustible × 965.7.		
2. Loss due to moisture in coal = per cent of moisture referred to combustible ÷ 100 × $[(212 - t) + 966 + 0.48\,(T - 212)]$ (t = temperature of air in the boiler-room, T = that of the flue gases).		
3. Loss due to moisture formed by the burning of hydrogen = per cent of hydrogen to combustible ÷ 100 × 9 × $[(212 - t) + 966 + 0.48\,(T - 212)]$.		
4.* Loss due to heat carried away in the dry chimney gases = weight of gas per pound of combustible × 0.24 × $(T - t)$.		
5.† Loss due to incomplete combustion of carbon = $\frac{CO}{CO_2 + CO} \times \frac{\text{per cent C in combustible}}{100} \times 10{,}150$.		
6 Loss due to unconsumed hydrogen and hydrocarbons, to heating the moisture in the air, to radiation, and unaccounted for. (Some of these losses may be separately itemized if data are obtained from which they may be calculated.)		
Totals		100 00

---

* The weight of gas per pound of carbon burned may be calculated from the gas analyses as follows:

Dry gas per pound carbon $= \frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)}$, in which CO_2, CO, O, and N are the percentages by volume of the several gases. As the sampling and analyses of the gases in the present state of the art are liable to considerable errors, the result of this calculation is usually only an approximate one The heat balance itself is also only approximate for this reason, as well as for the fact that it is not possible to determine accurately the percentage of unburned hydrogen or hydrocarbons in the flue gases.

The weight of dry gas per pound of combustible is found by multiplying the dry gas per pound of carbon by the percentage of carbon in the combustible, and dividing by 100.

† CO_2 and CO are respectively the percentage by volume of carbonic acid and carbonic oxide in the flue gases. The quantity of 10,150 = Number of heat units generated by burning to carbonic acid one pound of carbon contained in carbonic oxide.

XXIII. *Report of the Trial.* — The data and results should be reported in the manner given in either one of the two following tables, omitting lines where the tests have not been made as elaborately as provided for in such tables.* Additional lines may be added for data relating to the specific object of the test. The extra lines should be classified under the headings provided in the tables, and numbered as per preceding line, with subletters *a*, *b*, etc. The Short Form of Report, Table No. 2, is recommended for commercial tests and as a convenient form of abridging the longer form for publication when saving of space is desirable.† For elaborate trials, it is recommended that the full log of the trial be shown graphically, by means of a chart.

TABLE NO. 1.

DATA AND RESULTS OF EVAPORATIVE TEST.

Arranged in accordance with the Complete Form advised by the Boiler Test Committee of the American Society of Mechanical Engineers. Code of 1899.

Made by . . . . of. . . . . boiler at . . . . to determine . . . .

. . . .

Principal conditions governing the trial . . . .

. . . .

. . . .

Kind of fuel . . . .
Kind of furnace . . . .
State of the weather . . . .

Method of starting and stopping the test ("standard" or "alternate," Art. X and XI, Code, and note, p. 713) . . . .

1. *Date of trial* . . . .
2. *Duration of trial* . . . . hours.

Dimensions and Proportions.

A complete description of the boiler, and drawings of the same if of unusual type, should be given on an annexed sheet.

3. *Grate surface* . . . . *width* . . *length* . . . . . . *area* . . . . . sq. ft.
4. Height of furnace . . . . ins.
5. Approximate width of air spaces in grate . . . . in.
6. Proportion of air space to whole grate surface . . . . per cent.

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* The items printed in italics correspond to the items in the "Short Form of Code."

† See Forms for recording the observations, pages 708 and 709.

7. *Water-heating surface* . . . . sq. ft.
8. *Superheating surface* . . . . "
9. Ratio of water-heating surface to grate surface . . . . — to 1.
10. Ratio of minimum draft area to grate surface . . . . 1 to —

Average Pressures.

11. *Steam pressure by gauge* . . . . lbs. per sq. in.
12. *Force of draft between damper and boiler* . . . . ins. of water.
13. Force of draft in furnace . . . . " "
14. Force of draft or blast in ashpit . . . . " "

Average Temperatures.

15. Of external air . . . . deg.
16. Of fireroom . . . . "
17. Of steam . . . . "
18. Of feed water entering heater . . . . "
19. Of feed water entering economizer . . . . "
20. *Of feed water entering boiler* . . . . "
21. *Of escaping gases from boiler* . . . . "
22. Of escaping gases from economizer . . . . "

Fuel.

23. Size and condition . . . .
24. Weight of wood used in lighting fire . . . . lbs.
25. *Weight of coal as fired* * . . . . "
26. *Percentage of moisture in coal* † . . . . per cent.
27. *Total weight of dry coal consumed* . . . . lbs.
28. *Total ash and refuse* . . . . "
29. Quality of ash and refuse . . . .
30. Total combustible consumed . . . . lbs.
31. *Percentage of ash and refuse in dry coal* . . . . per cent.

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* Including equivalent of wood used in lighting the fire, not including unburnt coal withdrawn from furnace at times of cleaning and at end of test. One pound of wood is taken to be equal to 0.4 pound of coal, or, in case greater accuracy is desired, as having a heat value equivalent to the evaporation of 6 pounds of water from and at 212 degrees per pound. (6 × 970.4 = 5822 B.t.u.) The term "as fired" means in its actual condition, including moisture.

† This is the total moisture in the coal as found by drying it artificially, as described in Art. XV of Code. (Note, p. 717.)

Proximate Analysis of Coal.

	Of Coal.	Of Combustible.
32. Fixed carbon	per cent.	per cent.
33. Volatile matter	"	"
34. Moisture	"	—
35. Ash	"	—
	100 per cent.	100 per cent.
36. Sulphur, separately determined	per cent.	per cent.

Ultimate Analysis of Dry Coal.

(Art. XVII, Code.)

	Of Coal.	Of Combustible
37. Carbon (C)	per cent.	per cent.
38. Hydrogen (H)	"	"
39. Oxygen (O)	"	"
40. Nitrogen (N)	"	"
41. Sulphur (S)	"	"
42. Ash	"	—
	100 per cent.	100 per cent.
43. Moisture in sample of coal as received	per cent.	per cent

Analysis of Ash and Refuse.

44. Carbon	per cent.
45. Earthy matter	"

Fuel per Hour.

46. *Dry coal consumed per hour*	lbs.
47. Combustible consumed per hour	"
48. *Dry coal per square foot of grate surface per hour*	"
49. Combustible per square foot of water-heating surface per hour	"

Calorific Value of Fuel.

(Art. XVII, Code.)

50. *Calorific value by oxygen calorimeter, per lb. of dry coal*	B.t.u.
51. *Calorific value by oxygen calorimeter, per lb. of combustible*	"
52. Calorific value by analysis, per lb. of dry coal *	"
53. Calorific value by analysis, per lb. of combustible	"

Quality of Steam

54. *Percentage of moisture in steam*	per cent.
55. *Number of degrees of superheating*	deg.
56. Quality of steam (dry steam = unity)	

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* See formula for calorific value under Article XVII of Code, and note, p. 719.

Water.

57. *Total weight of water fed to boiler** lbs.
58. Equivalent water fed to boiler from and at 212 degrees "
59. *Water actually evaporated, corrected for quality of steam* "
60. Factor of evaporation † "
61. Equivalent water evaporated into dry steam from and at lbs.
212 degrees.‡ (Item 59 × Item 60.) "

Water per Hour.

62. *Water evaporated per hour, corrected for quality of steam* "
63. *Equivalent evaporation per hour from and at 212 degrees*‡ "
64. *Equivalent evaporation per hour from and at 212 degrees per square foot of water-heating surface*‡ "

Horse Power.

65. *Horse power developed.* (34½ *lbs. of water evaporated per hour into dry steam from and at* 212 *degrees, equals one horse power.*)§ H.P.
66. *Builders' rated horse power* "
67. *Percentage of builders' rated horse power developed* per cent.

Economic Results.

68. *Water apparently evaporated under actual conditions per pound of coal as fired.* (*Item* 57 ÷ *Item* 25.) lbs.
69. *Equivalent evaporation from and at* 212 *degrees per pound of coal as fired.*‡ (*Item* 61 ÷ *Item* 25.) "
70. *Equivalent evaporation from and at* 212 *degrees per pound of dry coal.*‡ (*Item* 61 ÷ *Item* 27.) "
71. *Equivalent evaporation from and at* 212 *degrees per pound of combustible.*‡ (*Item* 61 ÷ *Item* 30.) "

(If the equivalent evaporation, Items 69, 70, and 71, is not corrected for the quality of steam, the fact should be stated.)

---

* Corrected for inequality of water level and of steam pressure at beginning and end of test.

† Factor of evaporation $= \frac{H - h}{965.7}$, in which H and h are respectively the total heat in steam of the average observed pressure, and in water of the average observed temperature of the feed. (See note, p. 722, for a discussion of Items 60 and 61.)

‡ The symbol "U.E.," meaning "Units of Evaporation," may be conveniently substituted for the expression "Equivalent water evaporated into dry steam from and at 212 degrees," its definition being given in a footnote.

§ Held to be the equivalent of 30 pounds of water per hour evaporated from 100° F. into dry steam at 70 pounds gauge pressure.

Efficiency.

(Art. XXI, Code, and note, p. 719.)

72. *Efficiency of the boiler; heat absorbed by the boiler per pound of combustible divided by the heat value of one pound of combustible* *.......... per cent.
73. *Efficiency of boiler, including the grate; heat absorbed by the boiler, per pound of dry coal, divided by the heat value of one pound of dry coal*.......... "

Cost of Evaporation.

74. *Cost of coal per ton of —— pounds delivered in boiler-room*.. $
75. Cost of fuel for evaporating 1000 lbs. of water under observed conditions.......... $
76. *Cost of fuel for evaporating* 1000 *lbs. of water from and at* 212 *degrees* .... $

Smoke Observations.

77. Percentage of smoke as observed.......... per cent.
78. Weight of soot per hour obtained from smoke meter .. . ounces.
79. Volume of soot per hour obtained from smoke meter..... cub. in.

Methods of Firing.

80. Kind of firing (spreading, alternate, or coking) ......
81. Average thickness of fire ....
82. Average intervals between firings for each furnace during time when fires are in normal condition ....
83. Average interval between times of levelling or breaking up.

Analyses of the Dry Gases.

84. Carbon dioxide (CO_2).......... per cent.
85. Oxygen (O).......... "
86. Carbon monoxide (CO).......... "
87. Hydrogen and hydrocarbons.......... "
88. Nitrogen (by difference) (N).......... "

100 per cent.

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* In all cases where the word combustible is used, it means the coal without moisture and ash, but including all other constituents. It is the same as what is called in Europe "coal dry and free from ash."

TABLE NO. 2.

DATA AND RESULTS OF EVAPORATIVE TEST.

Arranged in accordance with the Short Form advised by the Boiler Test Committee of the American Society of Mechanical Engineers. Code of 1899.

Made by...............on.............boiler, at....................to determine..
Kind of fuel...
Kind of furnace..

Method of starting and stopping the test ("standard" or "alternate," Art. X and XI, Code)...................................

Grate surface.. sq. ft.
Water-heating surface.................................. "
Superheating surface.................................. "

Total Quantities.

1. Date of trial ....
2. Duration of trial .... hours.
3. Weight of coal as fired * .... lbs.
4. Percentage of moisture in coal * .... per cent.
5. Total weight of dry coal consumed.................. lbs.
6. Total ash and refuse .... "
7. Percentage of ash and refuse in dry coal .... per cent.
8. Total weight of water fed to the boiler *.......... lbs.
9. Water actually evaporated, corrected for moisture or superheat in steam.................................... "
10. Equivalent water evaporated into dry steam from and at 212 degrees*.. "

Hourly Quantities.

11. Dry coal consumed per hour......................... lbs.
12. Dry coal per square foot of grate surface per hour .... "
13. Water evaporated per hour corrected for quality of steam . "
14. Equivalent evaporation per hour from and at 212 degrees *. "
15. Equivalent evaporation per hour from and at 212 degrees per square foot of water-heating surface *............. "

* See footnotes of Complete Form.

Average Pressures, Temperatures, etc.

16. Steam pressure by gauge.... .... lbs. per sq. in.
17. Temperature of feed water entering boiler .. .. deg.
18. Temperature of escaping gases from boiler... ...... "
19. Force of draft between damper and boiler .... ins. of water.
20. Percentage of moisture in steam, or number of degrees of superheating. . ...per cent or deg.

Horse Power.

21. Horse power developed. (Item 14 ÷ 34½.) *............ H.P.
22. Builders' rated horse power . . .. "
23. Percentage of builders' rated horse power developed .. .. per cent.

Economic Results.

24. Water apparently evaporated under actual conditions per pound of coal as fired. (Item 8 ÷ Item 3.).. . lbs.
25. Equivalent evaporation from and at 212 degrees per pound of coal as fired.* (Item 10 ÷ Item 3.). . . . . "
26. Equivalent evaporation from and at 212 degrees per pound of dry coal.* (Item 10 ÷ Item 5.). . ... "
27. Equivalent evaporation from and at 212 degrees per pound of combustible.* [Item 10 ÷ (Item 5 − Item 6.)]. .. "

    (If Items 25, 26, and 27 are not corrected for quality of steam, the fact should be stated)

Efficiency.

28. Calorific value of the dry coal per pound . ... B.t.u.
29. Calorific value of the combustible per pound .... "
30. Efficiency of boiler (based on combustible) * .... per cent.
31. Efficiency of boiler, including grate (based on dry coal)... "

Cost of Evaporation.

32. Cost of coal per ton of —— lbs. delivered in boiler-room . $
33. Cost of coal required for evaporating 1000 pounds of water from and at 212 degrees............... .... $

* See footnotes of Complete Forms.

DEPARTMENT OF EXPERIMENTAL ENGINEERING — SIBLEY COLLEGE.

COAL AND WATER LOG OF BOILER TEST.

Date of Test..
Kind of Boiler..
Condition of Boiler..

Kind of Fuel..............................
Condition of Piping..............................
....

OBSERVERS:
Fuel..............................
Water..............................
Refuse..............................

Time.	Feed Water.			Fuel.				Refuse.				Fire.		Water Level.		Remarks.
	Weight.			Time.	Weight.			Time.	Weight.			Time.	Condition.	Time.	Gauge.	
	Gross.	Tare.	Net.		Gross.	Tare.	Net.		Gross.	Tare.	Net.					

DEPARTMENT OF EXPERIMENTAL ENGINEERING — SIBLEY COLLEGE.

LOG OF BOILER TEST.

Made by .... *Date* ....

Kind of Boiler .... *Mfg. by* ....

		Pressures.			Temperatures.							Water Gauge.	
No.	Time.	Barometer.	Steam Gauge.	Draft Gauge.	External Air.	Boiler-Room.	Flue.	Feed Water.	Calorimeter.	Steam.	Furnace.	From Initial.	
1.													
2.													
3.													
4.													
5.													
6.													
7.													
8.													
9.													
10.													
11.													
12.													
13.													
14.													
15.													
16.													
17.													
18.													
19.													
20.													
21.													
22.													
23.													

360. Notes on the Rules. — *Rule I.* Appendix II states the possible objects of the test at greater length, and is quoted in full.

> In preparing for and conducting trials of steam boilers, the special object of the proposed trial should be clearly defined and steadily kept in view.
>
> 1. If it be to determine the efficiency of a given style of boiler or of boiler setting under normal conditions, the boiler, brickwork, grates, dampers, flues, pipes, in short, the whole apparatus, should be carefully examined and accurately described, and any variation from a normal condition should be remedied if possible, and if irremediable, clearly described and pointed out.
>
> 2. If it be to ascertain the condition of a given boiler or set of boilers with a view to the improvement of whatever may be faulty, the conditions actually existing should be accurately observed and clearly described.
>
> 3. If the object be to determine the relative value of two or more kinds of coal, or the actual value of any kind, exact equality of conditions should be maintained if possible, or, where that is not practicable, all variations should be duly allowed for.
>
> 4. Only one variable should be allowed to enter into the problem; or, since the entire exclusion of disturbing variations cannot usually be effected, they should be kept as closely as possible within narrow limits, and allowed for with all possible accuracy.

Rule II. Appendix X specifies in detail the dimensions and data that should be taken and is quoted in full.

> The report should include a complete description of the boiler, which, for special boilers, should be written out at length, but generally can conveniently be presented in tabular form substantially as follows:
>
> Type of boiler.
> Diameter of shell.
> Length of shell.
> Number of tubes.
> Diameter of tubes.
> Length of tubes.
> Diameter of steam drum.
> Width of furnace.
> Length of furnace.
> Kind of grate bars.
> Width of air spaces (in grate bars).
> Ratio of area of grate to area of air spaces.
> Area of chimney.
> Height of chimney.
> Length of flues connecting to chimney.

Area of flues connecting to chimney.
Grate surface.
Heating surface $\left\{\begin{array}{l}\text{Water}\\ \text{Steam}\\ \text{Total}\end{array}\right.$
Area of draft through or between tubes.
Ratio grate to heating surface.
Ratio draft area to grate.
Ratio draft area to total heating surface.
Water space.
Steam space.
Ratio grate to water space.
Ratio grate to steam space.

Rule IV. It is an easy matter to test, with the same kind of coal, *different* boilers which are to be compared in regard to efficiency or capacity. All that would appear to be necessary to specify the coal is that it should be a good grade of its kind. To attempt to say what is "commercially standard" coal might lead to difficulties.

In most practical cases a buyer of boilers would naturally desire to use the coal most common in his neighborhood, that being the kind which he can obtain at least cost, and will certainly call for guarantees of efficiency or capacity based upon this coal. This fact at once disposes of the matter of choice of coal. In order to prevent misunderstandings and possible litigation, the kind of coal should be specified, and if possible the proximate analysis, including the *heat value* of the coal, should be distinctly specified in every boiler guarantee.

Rule V.—For details concerning measurement of water and calibration of apparatus, see Chapters VI, VII, and XII.

Rule VII. The provision requiring the disconnecting or blank flanging of all water connections and blow-off pipes, except the pipe through which the boiler receives its feed water, should be followed to the letter whenever possible. There are now and then cases in practice, however, where, under the circumstances existing, this is not possible. There are also cases in which it is desired to test a boiler plant in the then existing conditions and where the determination of the losses through leaky blow-off cocks, leaky tubes, etc., is a part of the test. In such cases the amount of such loss can be determined by making what is called a " leakage test " at the end of the main test.

To make this test, proceed as follows: Previous to the test, fasten behind each gauge glass a scale which can be read to at least .05 inch. Just after the main test is completed, close all of the outlets from the boiler or battery through feed water and steam connections, and have the pressure maintained constant at the working pressure by proper manipulation of the drafts. Observe the water level for at least an hour, taking the readings at first, say every minute and later on every two or three minutes, depending upon how rapidly the water level sinks. Plot a curve between scale reading and time. This will usually be of the form shown in Fig. 489, the rapid drop from *a* to *b* being in part due to the natural shrinking of the volume of water as active ebullition ceases. The more nearly straight part *bc* represents true leakage. The leakage in the time represented by *b'c* will then be equal to *bb'* inches. Or *cb'* may be prolonged back to *a'*, in which case *a'a''* represents the inches of leakage per hour. What this amounts to in pounds depends upon the size of the steam drum and upon the

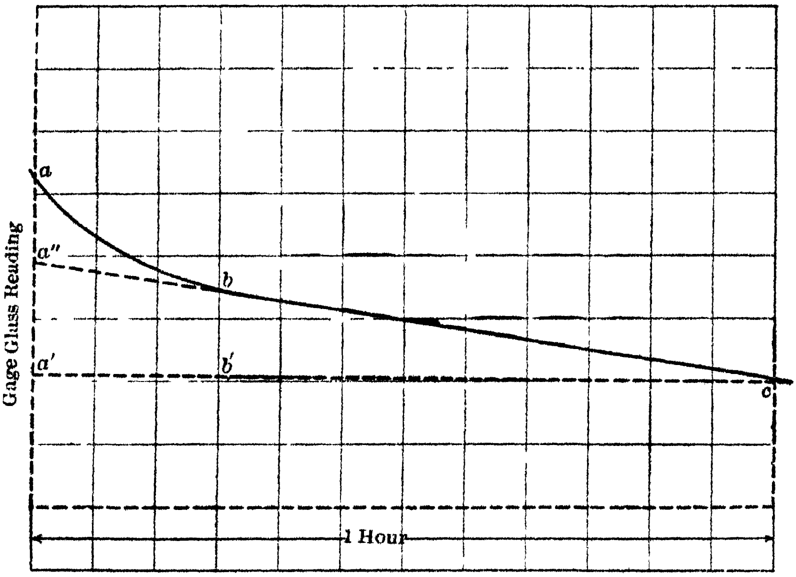

FIG. 489.—DETERMINATION OF BOILER LEAKAGE.

location of the gauge glass with reference to the center line of the drum. In computing the leakage in pounds, the temperature correction should not be forgotten, since the volume of water leaking out is at the temperature corresponding to the steam pressure.

In guarantee tests the leakage correction should not amount to more than one or two per cent of the total water evaporated. If it does, one should either be absolutely certain of this correction, or the boiler should be put in proper shape and retested.

Rules IX, X, and *XI.* The " standard " method of starting and stopping a test, although defended by some engineers, is strongly criticized by the majority, and the "alternate " method is to-day the one most used. Professor Wm. Kent, in Appendix XXXVI to the Code, briefly compares the two methods and also warns against some possible errors in the use of the alternate method, both with regard to the condition of the fire and the influence of the latter upon the movements of the apparent water level.

Of the two methods of starting and stopping a test, the so-called " standard " method and the " alternate " method, the writer prefers the latter, believing that the errors in the estimation of the quantity and condition of the small amount of coal left on the grate after cleaning are less than the errors of the " standard " method, which are due first, to cooling of the boiler at the beginning and end of the test; second, to the imperfect combustion of the fuel at the beginning; and third, to excessive air supply through the thin fire while burning down before the end of the test.

A special caution is needed against a modification of the " alternate " method, which has been adopted by some testing engineers within the past few years. It consists in taking the starting and the stopping times each at a time subsequent to the cleaning, say after 400 pounds of coal has been fired since the cleaning. There are two sources of serious error in this method, one causing an incorrect measurement of the coal, the other an incorrect measurement of the water. Suppose 200 pounds of hot coke are left on the grate at the end of cleaning and 400 pounds of fresh coal are added by the end of, say, half an hour after cleaning. If the coal left at the end of the cleaning, and the boiler walls also, are very hot, and the coal is highly volatile and dry, and the pieces of such size as not to choke the air supply, the fire may burn so briskly that at the end of the half hour the fuel value of the partly burned coal left out of the total 600 pounds is equivalent only to 200 pounds of coal. If, on the contrary, the hot coke on the grates at the end of the cleaning, and the boiler walls, are considerably cooled, if the fresh coal fired is moist and of small size, such as the slack of run-of-mine bituminous coal, which is often found in one portion of a pile in greater quantity than in another, the fire during the half hour may burn so sluggishly that the coal and coke on the grate at the end of the half hour may have a fuel value equal to 400 pounds of coal. If, in this case, it is assumed that the quantity and condition of the coal at the end of the half hour after cleaning are the same at the starting and stopping time, and if the fire burned briskly during the half hour before starting and slowly during the half hour before stopping, the boiler will be charged with more coal than was actually burned. If, on the contrary, the coal burns away more slowly during the half hour after the cleaning before the starting time and more rapidly during the half hour before the end of the test, the boiler is not charged with as much coal as was actually burned.

The error in water measurement is due to the fact that the condition of the fire, and especially the quantity of flaming gases arising from it, influences the height of the water level. A bright, hot fire, or a fire with an abundance of burning gas proceeding from it, causes the water level to rise; while anything that cools the furnace, such as freshly fired coal, an open fire door, or a check to the draft, causes the water level to fall. A rise or a fall of several inches in a few seconds frequently occurs when bituminous coal is used. If the water level is noted at the starting of the test, when it is raised by a bright fire, and at the end of the test, when it is depressed by the stoppage of violent ebullition or of rapid circulation, due to the cooling of the fire, the boiler will be credited with more water than was really evaporated, and *vice versa*.

The only correct times to be noted as the starting and the stopping times are when the smallest amount of fuel is on the grate and when it is in the most burned-out condition; that is, just before firing fresh coal after cleaning, and when the water level is in its most quiet condition and the least raised by ebullition; that is, after the furnace door has been kept open for some time for cleaning and the furnace therefore is in its coolest state. This condition of fire and of water level can be duplicated immediately after cleaning the fire; but there is no certainty of duplication of any condition when there is a bright fire and consequent rapid steaming.

Rule XIII. In keeping the record and in presenting the results, it is sometimes of benefit to construct what is known as a log chart. Such a chart shows at a glance the conditions that obtained for the entire period of the test. An example of such a log is shown in Fig. 490, which is a reproduction of the one given by Mr. G. H. Barrus, in Appendix XXXVIII of the Code.

Concerning the methods of tallying, the number of assistants required, etc., Appendix IV, written by Mr. C. E. Emery, gives some valuable points.

It should be steadily kept in mind that the principal observations to be made are the quantities of coal consumed and of water evaporated. If these quantities are ascertained accurately, and the conditions made the same at the beginning and end of test, the most important requisites of a boiler trial will be secured. Other observations have their value both for scientific and practical purposes, but are in most cases subsidiary.

Boiler tests are often undertaken with insufficient apparatus and assistance It is possible for a single person to test one boiler, or even several in a battery, but it requires a great deal of labor to do so, and in many cases such persons would be so fatigued as to be liable to make a simple error vitiating the results. He would, moreover, at no time be able to give proper oversight to the test so as to prevent accidental or unauthorized interferences It is very desirable, in fact almost indispensable, that an assistant be detailed to weigh the coal, and another to weigh or measure the water; if calorimeter tests are to be undertaken, still another assistant should be provided The engineer in charge is then left free to oversee the work of all, and relieve either temporarily when necessary. Engineers are frequently called upon to make boiler trials in connection with parties whose interests are antagonistic to a fair test, and frequently the voluntary assistance of busybodies is likely to produce errors in the results. It is therefore essential to have trustworthy assistants, and those of sufficient caliber not to be confused by interested parties, who will frequently endeavor in the most plausible manner to make out that a certain measure of coal has already been tallied, or that a certain tank of water has not been tallied.

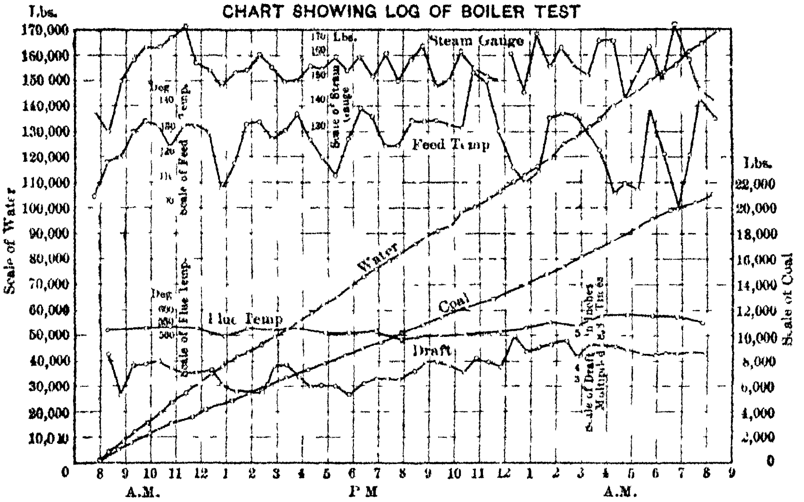

FIG 490

In the first engine trials at the American Institute Exhibition (1869), in the

Centennial trials (1876), and since in private trials respecting performance of boilers as between the contractor and purchaser, the writer has arranged for both interests to take the data at the same moment, with instructions, if agreement could not be had, that the difference be at once referred to him.

In weighing the coal, the barrow or vessel used should be balanced on a scale and then filled to a certain definite weight. The laborer will soon learn to fill a vessel to the same weight within a few pounds by counting the number of shovels thrown in, when the change of a lump or two to or from a small box alongside the scale will balance it.

[Mr. Emery next discusses methods of measuring water.]

A simple tally should never be trusted. Nothing seems more reliable to an inexperienced observer than to mark 1, 2, 3, 4 with a diagonal cross mark for 5; but when there are waits of several minutes between the marks, and several operations performed after the tally is made, there will be confusion in the mind whether or not the tally has actually been made. The tallies both of weights of coal and of tanks of water should be written on separate lines, the time noted opposite each, and the records always made at the beginning or termination of some particular operation; for instance, in weighing coal only at the time when barrel or bucket is dumped on the fire-room floor. It is desirable to have a number of coincident records of coal and water throughout the trial, so that in case of accident it may be held to have ended at one of such times. The uniformity of the operations may also be tested in this way from time to time. For this reason it will be found convenient to fire from a wheelbarrow set on a scale and to have a float or water-gauge connected with the tank from which the water is pumped; by which means the coal and water used may, in an evident way, be ascertained for any desired interval.

Rules IX, X, XI, XII, and XIII. If uniformity of operation of the boiler and furnace is secured, as it should be (Rule XII), it is obvious that the curves of " total water " and " total coal " against time (curves marked "water" and "coal" in Fig. 490) will be straight lines. The slopes of these lines are the rates of evaporation of water and supply of fuel. The real object of a boiler test is to obtain the ratio of these rates. It is evident that these rates may be obtained without regard to the end conditions of the tests. It appears then that there is a third method of " starting and stopping " a boiler test, a method more scientific and more accurate than either the "standard" or the "alternate" methods.

This third method requires merely that the boiler be brought to, and kept at, uniform operation for the length of time over which the test is to continue. During this time observations should be made of the amounts of water and coal supplied during every ten or fifteen minute interval, and of the amount of refuse coming through the grates every half hour. The curves of " total water," " total coal," and " total refuse " against time are then to be plotted. The rates are to be found from the slopes of these curves, using those portions which are straight lines, and leaving out the curved end portions where the boiler was heating up or cooling off. In working up the test the other observations, as steam pressure, quality, feed temperature, flue temperature, etc., are to be averaged and used only for that period of time within which the coal, water, and refuse curves are straight lines. The " end conditions " need not be considered; their effects are eliminated by the determination of rates.

Rule XIV. For a discussion of the methods of determining moisture in steam, see Chapter XIV.

The specific heat of superheated steam has now been determined with sufficient accuracy to allow of satisfactory determination of the total heat of superheated steam, see Chapter XI. It is advocated in the Code and in Appendix XIX to determine the range of superheat by finding the normal or saturation temperature of the steam by a special experiment, taking a reading of the thermometer when the fires are dull and superheat has disappeared. This method certainly has the advantage of doing away with the calibration of the thermometer with its attendant difficulty of duplicating the conditions of immersion of bulb and stem, surrounding air temperature, etc. The average steam pressure, however, must be exactly duplicated and maintained during the experiment, which may not be an easy matter. With the error of the steam gauges known, direct reference to the steam table for the saturation temperature is now probably of sufficient accuracy.

Rule XV. Of the two methods given for the determination of moisture in coal, the first must be held distinctly unreliable and in most cases useless. In the first place, an approximate determination is not what is wanted. In the second place, considering the sources of error involved and the usual lack of facilities in boiler houses for making close weighings, grave doubt must usually exist concerning the degree of approximation.

Regarding the second method of determination, see Chapter XIII.

Rule XVI. It is perhaps not superfluous to point out that the correct determination of the " ash " (total ash and refuse in the tables below) made during the test is not so simple a matter as it would appear at first sight, especially if the alternate method of starting and stopping is used. The entire matter of course rests on the correct estimation of the condition of the fire at beginning and end of test, and this in turn is largely a matter of experience. The accurate determination of the ash weight is of particular importance where the efficiency of the furnace, which may be of special type, is at issue, or where a heat balance as nearly correct as possible must be established.

The coal analysis of course shows the absolute minimum of ash that can be obtained on a test. But since no grate is perfect, there should, in most cases, be more "ash" than the analysis shows, due to unburned carbon dropping through. It may, however, happen that the coal is of such a nature that, under the draft conditions existing, a part of it or of its ash may be carried along with the gases for some distance and deposited in the passes or in the combustion chamber. If this action is suspected, provision should be made for determining the weight and composition of material so carried over.

In establishing the heat balance it becomes necessary to determine the combustible still remaining in the ash. The best way of doing this is by analysis. The scheme sometimes used of subtracting the percentage of true ash shown by the coal on analysis from the percentage of ash and refuse in the tests and calling the difference the carbon lost in "ash," may or may not be correct, depending upon how accurately the " ash " was determined on the test. Provided the latter is correct, the results of the two methods for carbon in "ash" should be the same, and in that sense the carbon in "ash" as found from the analysis may be used as an indication of the degree of accuracy of the "ash" determination on test.

Assuming that the combustible part of the refuse is carbon, there are two ways of determining this combustible. Either a chemical determination of the carbon is made by one of the usual methods, or the carbon may be burned out in the calorimeter, when the heat developed divided by the heating value of carbon should give the amount of carbon the sample contained. It may be difficult to keep the sample burning in the calorimeter, even in an atmosphere of oxygen, in which case the expedient of mixing the ash with a certain proportion of some known combustible may be used.

Rule XVII. The applicability of Du Long's or any similar formula to all grades of coal without any differentiation is a disputed question. It seems certain, however, that the method of computing heating values from analyses should be used only with great caution in the case of bituminous or semi-bituminous coals. See Chapter XIII for more detailed discussion of this matter, and of the making of coal analyses in general.

Rule XX. Condensation of the steam is not in this case an accurate method of obtaining the total heat imparted to the water.

Rules XXI, XXII, and the Tables. The subject matter of these rules is closely connected with the interpretation of the meaning of some of the items in the tables. It will therefore be best to consider them together.

The value 965.7 B.t.u. given in the Code (Rule XXI) for the heat of vaporization per pound of water from and at 212° F., has been changed to 970.4 B.t.u. by recent accurate researches, and this is the figure that should be employed.

The formula for " Efficiency of Boiler and Grate " in the same rule does not state whether coal as fired or dry coal is used as the basis of the computation. It is however shown, by Item 73 of Table No. 1, and Item 31 of Table No. 2, that the computation is to be based on dry coal.

Appendix XX of the Code states:

> " When the object of a boiler test is to determine its efficiency as an absorber of heat, or to compare it with other boilers, the efficiency based on combustible is the one which should be used; but when the object of the test is to determine the efficiency of the combination of the boiler, the furnace, and the grate, the efficiency based on coal must necessarily be used."

Every engineer will in general agree with the distinction made in this statement, but some disagreement of opinion exists whether *dry* coal or coal *as fired* should be used as the basis of computation for boiler and grate efficiency. If it is desired by this efficiency to characterize the performance of the boiler as a whole under the conditions of operation, it is certainly hard to see why dry coal should be used as a basis, since the coal is in general not dry and the assumption does not therefore conform to actual conditions, and, further, it is easily conceivable that the performance of the boiler and grate might have been quite different had dry coal been actually used. The efficiency based on combustible may serve as a basis of comparison for different boilers, but the efficiency of boiler and grate, the *real efficiency* of the apparatus, applies only to the particular boiler for which the computation is made, and there would therefore seem to be little reason for making an arbitrary assumption concerning any test condition.

The method of computing the boiler efficiency is not clearly enough defined. As the formula given is ordinarily applied, the term " combustible " does not represent quite the same basis above and below the line. The heat absorbed per pound of combustible would most certainly be computed from the equivalent evaporation per pound of combustible by multiplying the latter by 970.4 B.t.u. (Item 71 of Table No. 1.) But the " combustible " used in the derivation of Item 71 (Item 30) is computed from (coal as fired — refuse — moisture). The heating value of the combustible, on the other hand, is usually determined from calorimeter observations, taking "combustible " equal to (coal as burned — calorimeter ash — moisture). The difference lies in the fact that "refuse" and "calorimeter ash" are not the same, if fuel is lost through the grate, as it usually is to some extent. The heating value of each pound of combustible as determined under Item 30 is likely to be somewhat less than that given to the combustible on the basis of calorimeter computations, on account of the carbon lost, and the boiler efficiency is therefore generally stated too small, although the difference may not be great in most cases. The footnote to Item 72, defining combustible, is misleading if strictly interpreted as to "ash," because the equivalent evaporation per pound of combustible is not computed that way.

In some reports of tests, a third efficiency, that of the furnace, is separately computed. Unless some attempt is made to determine the distribution of the radiation losses between furnace and boiler proper, the *furnace efficiency* may be expressed as the ratio of boiler and grate efficiency to boiler efficiency.

The Code itself calls the *Heat Balance* given in Rule XXII "approximate," and this is certainly true concerning several of the items, notably Item 4. The heat balance given is based upon combustible, but it is much more common to base it upon the coal as fired, because the method gives a much clearer insight into what becomes of the heat in the coal supplied to the boiler. Further, such a heat balance is more rational in that it considers the boiler as a whole and thus analyzes the performance not only of the steam generator itself but also of the furnace, which is naturally what is desired in most cases.

A complete heat balance should state the following items:

1. Heat loss through incomplete combustion.
    (*a*) Loss of fuel through grate and over bridge wall.
    (*b*) Loss of combustible in chimney gases.
2. Loss in sensible heat in the flue gas, except that due to moisture.
3. Heat loss due to moisture in the flue gas.
    (*a*) Moisture in coal as fired.
    (*b*) Moisture formed from combustion of hydrogen in fuel.
    (*c*) Moisture as humidity in air used.
4. Heat losses due to radiation and other losses unaccounted for.
5. Heat absorbed in steam.

For a complete discussion of the method of computing the heat quantities for Items 1 to 3 inclusive, see pp. 531 to 538, Chapter XIII. Item 4 is obtained by the difference between the heating value of the fuel as fired and the sum of the rest of the items. Item 5 must be accurately computed, and is in all cases equal to ($xr + q - q$ of feed) B.t.u. times the number of pounds of water evaporated per pound of fuel as fired.

The items of Table No. 1 are in most cases self-explanatory in the light of the provisions of the Code. But certain of them call for explanation or comment.

Items 30, 51, and 53. It has already been pointed out that the "combustible" under 30 is somewhat different in its nature from that determined in either the proximate or chemical analyses, on account of the loss of fuel in refuse, and that the heating values given under 51 and 53 do not therefore strictly apply to the combustible as determined under 30.

Item 59 is obtained from 57 by multiplying the latter by the quality of steam (Item 54). This therefore represents the actual weight of *dry and saturated steam* generated.

Item 60. The formula given for the " Factor of Evaporation " is not thermally exact, irrespective of the fact that the denominator should be 970.4.

The form given is

$$E = \frac{H - h}{970.4}, \qquad \text{(I)}$$

in which H ($= \lambda$) is the total heat above 32 degrees in dry and saturated steam at the observed pressure, and h = the heat in the feed water above 32 degrees $= t - 32$, where t is the feed-water temperature.

This form neglects the heat carried away from the boiler in the moisture in steam and gives the boiler credit only for the heat that can actually be accounted for in the dry and saturated steam. This method of computation may be satisfactory if the boiler is considered merely as an instrument for generating dry and saturated steam, but considered as an apparatus transferring heat, the basis of computation is not correct. The *thermally* correct method would lead to a different result for the " Equivalent Evaporation from and at 212°." It is not contended that the method of computation used in the Code is not satisfactory, considering the boiler merely from the standpoint of its ability to generate dry steam, but it should be made clear that the method does not account for all the heat absorbed by the boiler shell.

The true factor of evaporation is

$$E = \frac{xr + q - t + 32}{970.4}, \qquad \text{(II)}$$

where the numerator expresses the heat above feed-water temperature in a pound of steam as actually made. 970.4 is the expenditure of heat that would have been necessary to generate one pound of dry steam if the feed-water temperature had been 212 degrees and the temperature of the steam made finally also 212 degrees (hence the term "from and at 212 degrees"). Hence the factor expresses the number of pounds of steam that could have been made from and at 212 degrees with the expenditure of heat applied to a pound of the boiler steam as actually made. It will be seen that the use of such a factor established a common basis (a heat unit of large size) for all evaporative tests and that we are therefore able directly to compare coals as to their evaporative qualities.

Item 61. It was pointed out above that a different result may be obtained for this item, depending upon which factor of evaporation is employed. To show the difference between the two methods on a heat basis, let

W = weight of water apparently evaporated (Item 57).
x = quality of steam (Item 54).
E_1 = factor of evaporation, form I, given in Code.
E_2 = factor of evaporation, form II above.

Then, as per Code,

$$\text{Equivalent Evaporation} = W \times E_1 = W \times \frac{\lambda - t + 32}{970.4}.$$

The thermally exact method would give

$$\text{Equivalent Evaporation} = W \cdot \frac{xr + q - t + 32}{970.4}.$$

The difference in the results is perhaps best illustrated by an average example. For an absolute steam pressure of 150 pounds, a feed-water temperature of 70 degrees and a quality of steam equal to 98 per cent, the second formula gives a result ½ per cent higher than the first, and consequently all of the "Economic Results" and the "Efficiencies" of the table will be higher by the same amount. For greater moisture in the steam the difference will be greater; thus if $x = .94$, with the same steam pressure and feed-water temperature, the difference is 1.6 per cent. For dry and saturated steam the formulas of course give identical results.

For superheated steam the second form of the equation for factor of evaporation will read,

$$E_2 = \frac{\lambda + C_p(T_2 - T_1) - t + 32}{970.4},$$

in which T_2 = temperature of the superheated steam
T_1 = temperature of the saturated steam at the same pressure, and
C_p = mean specific heat in the range from T_1 to T_2.

CHAPTER XVIII.

THE TESTING OF STEAM ENGINES, PUMPING ENGINES, AND LOCOMOTIVES.

THE American Society of Mechanical Engineers has established and adopted codes for the testing of steam engines, pumping engines, and locomotives. For the full wording of these codes the reader is referred to the Transactions of the Society.*

These codes in general not only discuss the objects of the various tests, and the methods of making the tests, but also go into the matter of instruments to be used, their choice, application, calibration, etc., besides giving methods of computing and presenting the results. In many instances these things have been thoroughly covered, sometimes at greater length, in previous chapters of this book, and to repeat these parts of the code text would therefore lead to unnecessary repetition. Wherever possible such parts have therefore been omitted and proper page references to other parts of the book are given. Additional comment upon some of the provisions is given in footnotes.

361. The Testing of Steam Engines. General Considerations and Definitions. - The Code concerns itself very largely with directions for complete plant tests. The testing of an engine alone is a comparatively simple matter. The test may be made to determine:

(*a*) Economy.
(*b*) Capacity.
(*c*) Mechanical Efficiency.
(*d*) Regulation.

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* Final Report of the Committee appointed to Standardize a System of Testing Steam Engines, Vol. XXIV, 1903.

Report of Committee on a Standard Method of Conducting Duty Trials of Pumping Engines, Vol. XI, 1890.

Report of the Committee on a Standard Method of Conducting Locomotive Tests, Vol. XIV, 1893

The simple *economy* test based on I.H.P. requires nothing further than the determination of the amount of steam used, the quality of the steam, the taking of indicator cards, and the recording of the average speed. In such a test it is immaterial by what means the engine is loaded, as long as any given load remains constant for a sufficient length of time. The steam consumption may be determined either from the boiler end or by means of the surface condenser, as the Code provides. The latter method is preferred because of greater accuracy.

Economy guarantees are sometimes made for other than the rated capacity. Thus many guarantees cover the steam consumption at ½ load, full load, and at some overload, as may be agreed upon. In the majority of these cases the guarantees are based upon developed, and not upon indicated, horse power in order to eliminate the mechanical efficiency of the engine. At the same time, separate agreements are made concerning the mechanical efficiency at the different loads. This kind of test requires means for loading the engine, such as brakes, generators, etc. In case electric generators are used to produce the load, it becomes necessary to correct for the efficiency of these machines in order to determine the net shaft (developed) horse power, unless the guarantees should happen to be based upon electrical horse power at the switchboard.

An economy test for a series of loads, as above described, is really also a test for *capacity* and for *mechanical efficiency.* During the progress of such a test a curve should be plotted between horse power and total steam used per hour. It will be found, if the work is accurate, that this curve is usually a straight line up to heavy overloads on an engine. This line is a graphical expression of *Willan's law* and is known as *Willan's line.* It serves as a check upon the accuracy of the test results as the test proceeds, and the tests may be considered completed if the highest load in a series gives a total steam consumption which is markedly off the line of the previous results.

The report on an economy test should include three curves: Willan's line, the relation between load and steam consumption per I.H.P. or per D.H.P. per hour (water-rate), and the relation between load and mechanical efficiency. Figure 491 shows a typical example of these curves for an automatic engine.

Regulation tests are usually not a part of the regular economy test. It is true that testing an engine under a series of loads will usually reveal that there is a certain amount of change in the average speed from load to load, the engine running a few turns per minute faster under low than under high loads, but that is no certain indication of the regulation. To obtain a reliable test for regulation, means should be provided to record the speed continuously. The test is then made by suddenly throwing load on or off. Figure 492 shows the results obtained on a regulation test of a 12″ × 18″ × 10″ tandem compound engine, the load variation being 125 H.P. Time and revolution records were taken by means of a chronograph. The curve shows that the engine was running at 280.5 R.P.M. under a load of 125 H.P. and at 283.5 R.P.M. at no load. The mean speed is 282 R.P.M., the speed variation is 3 R.P.M. The percentage of total variation, based on mean speed, is therefore $\frac{3.0}{282} = 1.06$ per cent, which corresponds to a regulation of .53 per cent from mean speed. The observations made also showed, taking the test when the load was thrown off, for instance, that it took 6 revolutions for

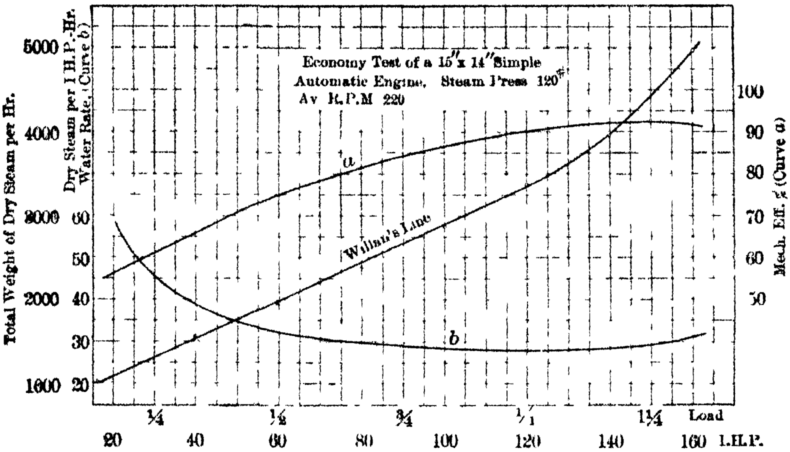

FIG. 491.—WATER-RATE AND MECHANICAL EFFICIENCY CURVES FOR A STEAM ENGINE

the engine to settle down to the mean speed, and further, that the revolution following that when the load was thrown off was made at the rate of 290 R.P.M., the second one following at the rate of 275.5 R.P.M. This is a total variation of 14.5 R.P.M. at a mean speed of 282.7, which amounts to 2.57 per cent above and below the mean.

The speed variation last discussed is largely a function of fly-wheel weight. In making regulation guarantees, both builder and purchaser should thoroughly understand what method of testing is to be employed and what method of computation is to be used, for it must be quite evident from the above that there is considerable chance for disagreement.* It might be said that the usual practice is to use the first method of computation, unless otherwise specified.

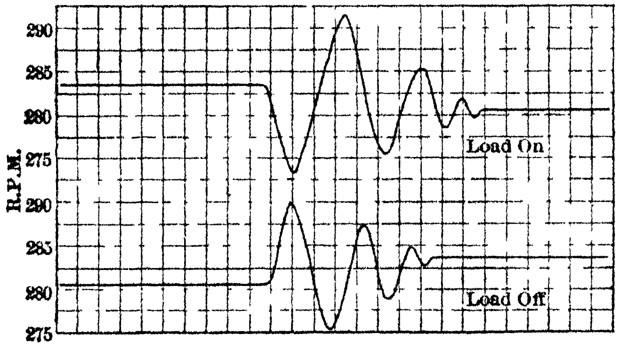

FIG. 492. — CURVES SHOWING SPEED REGULATION

Concerning *Efficiency Standards*, the Code defines some of them but does not mention all that are sometimes computed by engineers. The standards used may be defined as follows:

(*a*) *Thermal Efficiency.* — This is the ratio of the heat equivalent of the work done by the engine in a given time to the heat in the steam supplied in the same time to do that work. This efficiency may be computed on the I.H.P. or B.H.P. basis. In either case the heat equivalent in B.t.u. of the work is 2545 times the horse power, if the hour is used as the unit of time, and it is equal to 42.41 times the horse power if the minute is the time unit. The heat supplied is in every case equal to the weight of the steam supplied the engine in the time unit chosen, multiplied by the total heat content above 32 degrees in each pound of that steam. If the engine is using jackets, the heat in the jacket steam must be added to that in the engine steam. Whether the heat in the steam used by the engine auxiliaries should also be added depends upon contract agreement. *Note that no correction is made for the heat that the steam contains as it leaves the engine, irrespective of whether any use is or is not made of this residual heat in any other part of the plant.* (See Rule XXI of the Code.)

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* In general, for direct power work, as for the operation of shops, for instance, the speed variation from no load to full load, after the engine has again settled down, is the regulation to be looked after, the momentary hunting while the governor is adjusting itself being of little or no importance For the operation of generators for lighting purposes, however, the instantaneous variation is of fully as great importance as the regulation between full load and no load and should be covered by guarantees

(*b*) *Thermodynamic* (*Carnot*) *Efficiency.* — This is the efficiency of the Carnot cycle operating between the same upper and lower limits of temperature in the real engine. It is expressed by $\frac{T_1 - T_2}{T_1}$, where T_1 = the upper and T_2 the lower temperature limit. (See Art. 185, p. 351.) Thus if $T_1 = (350 + 460)$ (for 135 lbs. abs. steam pressure) and $T_2 = (213 + 460)$ (for 15 lbs. absolute steam pressure), the thermodynamic efficiency would be

$$\frac{(350 + 460) - (213 + 460)}{350 + 460} = \frac{350 - 213}{810} = 16.9 \text{ per cent.}$$

This efficiency has no real significance as far as any real engine is concerned, since no real steam engine now operating is based upon the Carnot cycle even in theory. It merely shows the maximum possible efficiency that could be reached between the given temperature limits.

(*c*) *Cylinder Efficiency.* — The real steam engine uses a cycle which approaches the theoretical Clausius cycle. (See Art. 185, p. 352.) The performance of an ideal engine using the Clausius cycle between the same pressure limits is therefore a more rational standard by which to compare real engine performance. (See note to Rule XXIV of the Code. The Clausius cycle is there called the Rankine cycle.)

The *cylinder efficiency* may then be defined as the ratio between the number of heat units that would be required per I.H.P. per hour by the ideal engine using the Clausius cycle divided by the heat units actually required by the real engine per I.H.P. per hour as shown by test. (See the type example given in connection with Rule XXIV of the Code.)

(*d*) *Mechanical Efficiency.* — As already defined, this is equal to the work delivered at the shaft (B.H.P.) to the work done in the cylinder (I.H.P.).

(*e*) *Plant Efficiency.* — This is usually understood to mean the over-all efficiency of a plant from the fuel to the useful power developed. It is equal to the continuous product of the thermal efficiencies of the separate parts of the chain of machinery by which the energy of the fuel is converted into work. Thus in the case of a lighting plant, if the efficiency of the boiler and grate of the boiler plant is 70 per cent, the thermal efficiency of the engine on the I.H.P. basis is 15 per cent; the mechanical efficiency of the engine is 95 per cent, and that of the generator to the switchboard is 92 per cent; the plant efficiency will be

$$= .70 \times .15 \times .95 \times .92 = 9.15 \text{ per cent.}$$

If in any given plant a part of the waste heat, as, for instance, that from the engine, is recovered and returned to the boiler, due allowance must be made, and the plant efficiency is correspondingly increased.

(*f*) *Performance of a Perfect Engine.* — A perfect engine is here understood to be one which operates on the Carnot cycle. Its efficiency would be that computed as shown under (*b*) above. If we represent the thermodynamic efficiency by e, the heat units that would be required by such an engine per I.H.P. hour would evidently be

$$= \frac{2545^*}{e} = \frac{2545\,T_1}{T_1 - T_2}\text{ B.t.u.}$$

The greatest possible amount of heat that can be taken out of the steam furnished any engine is the difference between the total heat H that the steam contains at the upper pressure and the heat of the liquid q_{ex} at the lower pressure. Hence the *steam consumption of the perfect engine* per I.H.P.-hour will be expressed by

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* This figure $= \frac{33,000 \times 60}{778}$. If the later determinations are taken, which give the value of the work equivalent of the B.t.u. = 777.5 ft.-lbs., the factor becomes = 2547.

$$\frac{2545\, T_1}{(H - q_{ex})(T_1 - T_2)} \text{ pounds.}$$

This is sometimes used as a standard for real engine performance, but since it is a standard impossible even of approach in any real engine, it has little or no practical value.

Whenever possible a *heat balance* should be established for every economy test. For a compound engine fitted with jackets on cylinders and receiver, such a heat balance should show the following items:

I. High-pressure Cylinder.
  - (*a*) Heat received in cylinder steam.
  - (*b*) Heat received in jacket steam.
  - (*c*) Heat rejected in exhaust.
  - (*d*) Heat rejected in jacket condensation.
  - (*e*) Heat utilized (heat equivalent of I.H.P. for H.P. cylinder).
  - (*f*) Heat lost by radiation, etc.

II. Receiver.
  - (*a*) Heat received in exhaust steam from H.P. cylinder.
  - (*b*) Heat received in receiver jacket steam.
  - (*c*) Heat supplied to L.P. cylinder.
  - (*d*) Heat rejected in jacket condensation.
  - (*e*) Heat lost by radiation, etc.

III. Low-pressure Cylinder.
  - (*a*) Heat received from receiver (item II (*c*)).
  - (*b*) Heat received in jacket steam.
  - (*c*) Heat rejected in exhaust steam { Heat in condensing water plus Heat in condensed steam.
  - (*d*) Heat rejected in jacket condensation.
  - (*e*) Heat utilized (heat equivalent of I.H.P. for L.P. cylinder).
  - (*f*) Heat lost by radiation, etc.

The establishment of this heat balance requires the determination of the quality of steam between the high-pressure cylinder and the receiver, and also the quality of the steam leaving the receiver. Unless the engine is operated condensing, the quality of the exhaust steam in the low-pressure exhaust must be found in order to determine item III (c). The loss represented by item III (f) will include the radiation losses from condenser and piping in the case of a condensing engine. Concerning the receiver, if the qualities of the steam are determined close to the high-pressure exhaust and low-pressure admission, the radiation loss, if computed from $e = c - a - b + d$, includes the losses from all of the piping between the two cylinders.

362. Rules for Conducting Steam-Engine Tests, Code of 1902.—

I. Object of Test. — Ascertain at the outset the specific object of the test, whether it be to determine the fulfillment of a contract guarantee, to ascertain the highest economy obtainable, to find the working economy and defects under conditions as they exist, to ascertain the performance under special conditions, to determine the effect of changes in the conditions, or to find the performance of the entire boiler and engine plant, and prepare for the test accordingly.

No specific rules can be laid down regarding many of the preparations to be made for a test, so much depends upon the local conditions; and the matter is one which must be left mainly to the good sense, tact, judgment, and ingenuity of the party undertaking it. One guiding principle must ever be kept in mind; namely, to obtain data which shall be thoroughly reliable for the purposes in view. If questions of contract are to be settled, it is of the first importance that a clear understanding be had with all the parties to the contract as to the methods to be pursued — putting this understanding, if necessary, in writing — unless these are distinctly provided for in the contract itself. The preparations for the measurement of the feed water and of the various quantities of condensed water in the standard heat-unit test should be made in such a manner as to change as little as possible the working conditions and temperatures of the plant.

II. General Condition of the Plant. — Examine the engine and the entire plant concerned in the test; note its general condition and any points of design, construction, or operation which bear on the objects in view. Make a special examination of the valves and pistons for leakage by applying the working pressures with the engine at rest, and observe the quantity of steam, if any, blowing through per hour.

If the trial has for an object the determination of the highest efficiency obtainable, the valves and pistons must first be made tight, and all parts of the engine and its auxiliaries, and all other parts of the plant concerned, should be put in the best possible working condition.*

III. DIMENSIONS, ETC. — Measure or check the dimensions of the cylinders in any case, this being done when they are hot. If they are much worn, the average diameter should be determined. Measure also the clearance, which should be done, if possible, by filling the spaces with water previously measured, the piston being placed at the end of the stroke. If the clearance cannot be measured directly, it can be determined approximately from the working drawings of the cylinder.

Measure the dimensions of auxiliaries and accessories, also those of the boilers so far as concerned in attaining the objects. It is well to supplement these determinations with a sketch or sketches showing the general features and arrangement of the different parts of the plant.

To measure the clearance by actual test, the engine is carefully set on the center, with the piston at the end where the measurement is to be taken. Assuming, for example, a Corliss engine, the best method to pursue is to remove the steam valve so as to have access to the whole steam port, and then fill up the clearance space with water, which is poured into the open port through a funnel. The water is drawn from a receptacle containing a sufficient quantity, which has previously been measured. When the whole space, including the port, is completely filled, the quantity left is measured, and the difference shows the amount which has been poured in The measurement can be most easily made by weighing the water, and the corresponding volume determined by calculation, making proper allowance for temperature. The proportion required is the volume in cubic inches thus found, divided by the volume of the piston displacement, also in cubic inches, and the result expressed as a decimal. In this test care should be taken that no air is retained in the clearance space when it is being filled with water.

* The Code under this provision gives extended directions for determining piston and valve leakage, both qualitatively and quantitatively In the first test, valves and pistons are placed in the positions desired and steam is then turned on, observations of the steam leaking past being made by opening indicator cocks, or other vents that may be available. Another scheme is the so-called "time method," in which steam is turned on full and then turned off, watching the drop of pressure by means of an indicator. In a perfectly tight cylinder this drop would be due only to condensation and would be very slow. In any actual case the drop will be more rapid, depending upon the degree of leakage. It will be observed that the qualitative method of determining leakage is practically valueless, except as it may serve to distinguish between a fairly tight engine and one that leaks very badly. The quantitative determination, made by condensing the leakage in some manner, is of course more accurate, but even the quantity so determined cannot be used as a correction to consumption figures determined on a test, because in operation the pressure conditions are not at all the same. Hence for very accurate work with reference to best consumption and maximum efficiency, the requirement of Rule II calling for tight piston and valves must be followed if possible to the letter.

The only difficulty which arises in measuring the clearance in this way is that occurring when the exhaust valves and piston are not tight, so that, as the water is poured in, it flows away and is lost. If the leakage is serious, no satisfactory measurement can be made, and it is better to depend upon the volume calculated from the drawing. If not too serious, however, an allowance can be made by carefully observing the length of time consumed in pouring in the water; then, after a portion of the water has leaked out, fill up the space again, taking the time, and measuring the quantity thus added, determining in this way the rate at which the leakage occurs. Data will thus be obtained for the desired correction.

IV. COAL. — When the trial involves the complete plant, embracing boilers as well as engine, determine the character of coal to be used. The class, name of the mine, size, moisture, and quality of the coal should be stated in the report. It is desirable, for purposes of comparison, that the coal should be of some recognized standard quality for the locality where the plant is situated.*

For New England and that portion of the country east of the Allegheny Mountains, good anthracite egg coal, containing not over 10 per cent of ash and semibituminous Clearfield (Pa.), Cumberland (Md.), and Pocahontas (Va.) coals are thus regarded. West of the Allegheny Mountains, Pocahontas (Va.) and New River (W. Va.) semibituminous and Youghiogheny or Pittsburg bituminous coals are recognized as standards.

V. CALIBRATION OF INSTRUMENTS. – All instruments and apparatus should be calibrated and their reliability and accuracy verified by comparison with recognized standards. Such apparatus as is liable to change or become broken during a test, as gauges, indicator springs, and thermometers, should be calibrated before and after the test. The accuracy of scales should be verified by standard weights. When a water meter is used, special attention should be given to its calibration, verifying it both before and after the trial, and, if possible, during its progress, the conditions in regard to water pressure and rate of flow being made the same in the calibrations as exist throughout the trial.

(*a*) GAUGES. — See Arts. 95-97, p. 188 of this book.

(*b*) THERMOMETERS. — See Art. 107, p. 219 of this book.

(*c*) INDICATOR SPRINGS. — See Arts. 329, 330, p. 597 of this book.

(*d*) WATER METERS. — A good method of calibrating a water meter is the following, reference being made to Fig. 493:

Two tees *A* and *B* are placed in the feed pipe, and between them two valves *C* and *D*. The meter is connected between the outlets of the tees *A* and *B*.

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* See the Boiler Code for further directions concerning observations on coal. It was pointed out there that there is usually no choice concerning the quality of the coal, and that there is in most cases no object in calling for any other grade of coal than that regularly used in the plant under consideration.

The valves *E* and *F* are placed one on each side of the meter. When the meter is running, the valves *E* and *F* are opened, and the valves *C* and *D* are closed. Should an accident happen to the meter during the test, the valves *E* and *F* may be closed and the valves *C* and *D* opened, so as to allow the feed water to flow directly into the boiler. A small bleeder *G* is placed between the valves *C* and *D*. The valve *G* is opened when the valves *C* and *D* are closed, in order to make sure that there is no leakage. A gauge is attached at *H*. When the meter is tested, the valves *C*, *D*, and *F* are closed, and the valves *E* and *I* are opened. The water flows from the valve *I* to a tank placed on weighing scales. In testing the meter the feed pump is run at the normal speed, and the water leaving the meter is throttled at the valve *I* until the pressure shown by the gauge *H* is the same as that indicated when the meter is running under the normal conditions. The piping leading from the valve *I* to the tank is arranged with a swinging joint, consisting merely of a loosely fitting elbow, so that it can be readily turned into the tank or away from it. After the desired pressure and speed have been secured, the end of the pipe is swung into the tank the instant that the pointer of the meter is opposite some graduation mark on the dial, and the water continues to empty into the tank while any desired number of even cubic feet are discharged, after which the pipe is swung away from the tank. The tests should be made by starting and stopping at the same graduation mark on the meter dial, and continued until at least 10 or 20 cubic feet are discharged for one test. The water collected in the tank is then weighed.

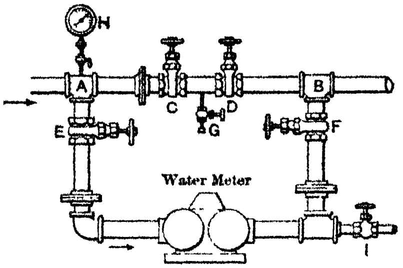

FIG. 493.—METHOD OF CONNECTING UP WATER METER FOR CALIBRATION.

The water passing the meter should always be under pressure in order that any air in the meter may be discharged through the vents provided for this purpose. Care should be taken that there is no air contained in the feed water. Should the feed-water pump draw from a hot-well, the height of the water in the hot-well must never be so low as the suction pipe of the pump. In case the speed of the feed pump cannot be regulated, as occurs in some cases where it is driven directly from the engine, a by-pass should be connected with the pipe leading from the pump, to allow some of the water to flow back into the hot-well, if the pump lowers the water in the hot-well beyond a given mark. The meter should be tested both before and after the engine trial, and several tests should be made of the meter in each case in order to obtain confirmative results. It is well to make preliminary tests to determine whether the meter works satisfactorily before connecting it up for an engine trial. The results should agree with each other for two widely different rates of flow.

VI. Leakages of Steam, Water, Etc. — In all tests except those of a complete plant made under conditions as they exist, the boiler and its connections, both steam and feed, as also the steam piping leading to the engine and its connections should, as far as possible, be made tight. If absolute tightness cannot be obtained (in point of fact it rarely can be) proper allowance should be made for such leakage in determining the steam actually consumed by the engine. This, however, is not required where a surface condenser is used and the water consumption is determined by measuring the discharge of the air pump. In such cases it is necessary to make sure that the condenser is tight, both before and after the test, against the entrance of circulating water, or if such occurs to make proper correction for it, determining it under the working difference of pressure. Should there be excessive leakage of the condenser it should be remedied before the test is made. When the steam consumption is determined by measuring the discharge of the air pump, any leakage about the valve or piston rods of the engine should be carefully guarded against

Make sure that there is no leakage at any of the connections with the apparatus provided for measuring and supplying the feed water which could affect the results. All connections should, so far as possible, be visible and be blanked off, and where this cannot be done, satisfactory assurance should be obtained that there is no leakage either in or out.

It is not always necessary to blank off a connecting pipe to make sure that there is no leakage through it. If satisfactory assurance can be had that there is no chance for leakage, this is sufficient. For example, where a straightway valve is used for cutting off a connecting pipe, and this valve has double seats with a hole in the bottom between them, this being provided with a plug or pet cock, assurance of the tightness of the valve when closed can be had by removing the plug or opening the cock Likewise, if there is a drain pipe beyond the valve, the fact that no water escapes here is sufficient evidence of the tightness of the valve. The main thing is to have positive evidence in regard to the tightness of the connections, such as may be obtained by the means suggested above; but where no positive evidence can be obtained, or where the leakage that occurs cannot be measured, it is of the utmost importance that the connections should be broken and blanked off.

Leakage of relief valves which are not tight, drips from traps, separators, etc., and leakage of tubes in the feed water-heater must all be guarded against or measured and allowed for.

It is well, as an additional precaution, to test the tightness of the feed-water pipes and apparatus concerned in the measurement of the water by running the pump at a slow speed for, say, fifteen minutes, having first shut the feed valves at the boilers. Leakage will be revealed by disappearance of water from the supply tank. In making this test, a gauge should be placed on the pump discharge in order to guard against undue or dangerous pressure.

(The Code next gives a "water-glass" method of determining boiler leakage, for which see p. 712 of this book.)

In making a test of an engine where the steam consumption is determined from the amount of water discharged from the surface condenser, leakage of the piston rods and valve rods should be guarded against; for if these are excessive, the test is of little use, as the leakage consists partly of steam that has already done work in the cylinder and of water condensed from the steam when in contact with the cylinder. If such leakage cannot be prevented, some allowance should be made for the quantity thus lost. The weight of water as shown at the condenser must be increased by the quantity allowed for this leakage.

VII. Duration of Test. — The duration of a test should depend largely upon its character and the objects in view. The standard heat test of an engine, and, likewise, a test for the simple determination of the feed-water consumption, should be continued for at least five hours, unless the class of service precludes a continuous run of so long duration. It is desirable to prolong the test the number of hours stated to obtain a number of consecutive hourly records as a guide in analyzing the reliability of the whole.

Where the water discharged from the surface condenser is measured for successive short intervals of time, and the rate is found to be uniform, the test may be of a much shorter duration than where the feed water is measured to the boiler. The longer the test with a given set of conditions, the more accurate the work, and no test should be so short that it cannot be divided into several intervals which will give results agreeing substantially with one another.

The commercial test of a complete plant, embracing boilers as well as engine, should continue at least one full day of twenty-four hours, whether the engine is in motion during the entire time or not. A continuous coal test of a boiler and engine should be of at least ten hours' duration, or the nearest multiple of the interval between times of cleaning fires.

VIII. Starting and Stopping a Test. — (*a*) *Standard Heat Test and Feed-water Test of Engine*. — The engine having been brought to the normal condition of running, and operated a sufficient length of time to be thoroughly heated in all its parts, and the measuring apparatus having been adjusted and set to work, the height of water in the gauge glasses of the boilers is observed, the depth of water in the reservoir from which the feed water is supplied is noted, the exact time of day is observed, and the test held to begin. Thereafter the measurements determined upon for the test are begun and carried forward until its close. If practicable, the test may be begun at some even hour or minute, but it is of the first importance to begin at such time as reliable observations of the water heights are obtained, whatever the exact time happens to be when these are satisfactorily determined. When the time for the close of the test arrives, the water should, if possible, be brought to the same height in the glasses and to the same depth in the feed-water reservoir as at the beginning, delaying the conclusion of the test if necessary to bring about this similarity of conditions. If differences occur, the proper corrections must be made.

Care should be taken in cases where the activity of combustion in the boiler furnaces affects the height of water in the gauge glasses that the same condition of fire and draughts are operating at one time as at the other. For this reason it is best to start and stop a test without interfering with the regularity of the operation of the feed pump, provided the latter may be regulated to run so as to supply the feed water at a uniform rate. In some cases where the supply of feed water is irregular, as, for example, where an injector is used of a larger capacity than is required, the supply of feed water should be temporarily shut off.

It is important to use great care in obtaining the average height of the water in the glasses, taking sufficient time to satisfactorily judge of the full extent of the fluctuation of the water line, and thereby its mean position. It is important, also, to refrain from blowing off the water column or its connecting pipes, either during the progress of the test or for a period of an hour or more prior to its beginning. Such blowing off changes the temperature of the water within, and thereby its specific gravity and height.

To mark the height of water in a gauge glass in a convenient way, a paper scale, mounted on wood and divided into tenths of inches, may be placed behind it or at its side.

(*b*) *Complete Engine and Boiler Test.* — For a continuous running test of combined engine or engines, and boiler or boilers, the same directions apply for beginning and ending the feed-water measurements as that just referred to under section (*a*). The time of beginning and ending such a test should be the regular time of cleaning fires, and the exact time of beginning and ending should be the time when the fires are fully cleaned, just preparatory to putting on fresh coal. In cases where there are a number of boilers, and it is inconvenient or undesirable to clean all fires at once, the time of beginning the test should be deferred until they are all cleaned and in a satisfactory state, all the fires being then burned down to a uniformly thin condition, the thickness and condition being estimated and the test begun just before firing the new coal previously weighed. The ending of the test is likewise deferred until the fires are all satisfactorily cleaned, being again burned down to the same uniformly thin condition as before, and the time of closing being taken just before replenishing the fires with new coal.

For a commercial test of a combined engine and boiler, whether the engine runs continuously for the full twenty-four hours of the day or only a portion of the time, the fires in the boilers being banked during the time when the engine is not in motion, the beginning and ending of the test should occur at the regular time of cleaning the fires, the method followed being that already given. In cases where the engine is not in continuous motion, as, for example, in textile mills, where the working time is ten or eleven hours out of the twenty-four, and the fires are cleaned and banked at the close of the day's work, the best time for starting and stopping a test is the time just before banking, when the fires are well burned down and the thickness and condition can be most satisfactorily judged. In these, as in all other cases noted, the test should be begun by observing the exact time, the thickness and condition of the fires on the grates, the height of water in the gauge glasses of the boilers, the depth of the water in the reservoir from which the feed water is supplied, and other conditions relating to the trial, the same observations being again taken at the end of the test, and the conditions in all respects being made as nearly as possible the same as at the beginning.

IX. MEASUREMENT OF HEAT UNITS CONSUMED BY THE ENGINE. — The measurement of the heat consumption requires the measurement of each supply of feed water to the boiler — that is, the water supplied by the main feed pump, that supplied by auxiliary pumps, such as jacket water, water from separators, drips, etc., and water supplied by gravity or other means; also the determination of the temperature of the water supplied from each source, together with the pressure and quality of the steam.

The temperatures at the various points should be those applying to the working conditions. The temperature of the feed water should be taken near the boiler. This causes the engine to suffer a disadvantage from the heat lost by radiation from the pipes which carry the water to the boiler, but it is nevertheless advisable on the score of simplicity. Such pipes would therefore be considered a portion of the engine plant. This conforms with the rule already recommended for the tests of pumping engines where the duty per million heat units is computed from the temperature of the feed water taken near the boiler. It frequently happens that the measurement of the water requires a change in the usual temperature of supply. For example, where the main supply is ordinarily drawn from a hot-well in which the temperature is, say, 100° F., it may be necessary, owing to the low level of the well, to take the supply from some source under a pressure or head sufficient to fill the weighing tanks used, and this supply may have a temperature much below that of the hot-well; possibly as low as 40° F. The temperature to be used is not the temperature of the water as weighed in this case, but that of the working temperature of the hot-well. The working temperature in cases like this must be determined by a special test, and included in the log sheets.

In determining the working temperatures, the preliminary or subsequent test should be continued a sufficient time to obtain uniform indications, and such as may be judged to be an average for the working conditions. In this test it is necessary to have some guide as to the quantity of work being done, and for this reason the power developed by the engine should be determined by obtaining a full set of diagrams at suitable intervals during the progress of the trial. Observations should also be made of all the gauges connected with the plant and of the water heights in the boilers, the latter being maintained at a uniform point so as to be sure that the rate of feeding during the test is not sensibly different from that of the main test.

The heat to be determined is that used by the entire engine equipment, embracing the main cylinders and all the auxiliary cylinders and mechanism concerned in the operation of the engine, including the air pump, circulating pump, and feed pumps, also the jacket and reheater when these are used. No deduction is to be made for steam used by auxiliaries unless these are shown by test to be unduly wasteful. In this matter an exception should be made in cases of guarantee tests where the engine contractor furnishes all the auxiliaries referred to. He should, in that case, be responsible for the whole, and no allowance should be made for inferior economy, if such exists. Should a deduction be made on account of the auxiliaries being unduly wasteful, the method of waste and its extent, as compared with the wastes of the main engine or other standard of known value, shall be reported definitely.

The steam pressure and the quality of the steam are to be taken at some point conveniently near the throttle valve. The quantity of steam used by the calorimeter must be determined and properly allowed for.

X. Measurement of Feed Water or Steam Consumption of Engine, Etc. — The method of determining the steam consumption applicable to all plants is to measure all the feed water supplied to the boilers, and deduct therefrom the water discharged by separators and drips, as also the water and steam which escape on account of leakage of the boiler and its pipe connections and leakage of the steam main and branches connecting the boiler and the engine. In plants where the engine exhausts into a surface condenser the steam consumption can be measured by determining the quantity of water discharged by the air pump, corrected for any leakage of the condenser, and adding thereto the steam used by jackets, reheaters, and auxiliaries as determined independently. If the leakage of the condenser is too large to satisfactorily allow for it, the condenser should, of course, be repaired and the leakage again determined before making the test.

(The Code here discusses methods of measuring water, for which see Chap. XII.)

The corrections or deductions to be made for leakage above referred to should be applied only to the standard heat-unit test and tests for determining simply the steam or feed-water consumption, and not to coal tests of combined engine and boiler equipment. In the latter, no correction should be made except for leakage of valves connecting with other engines and boilers, or for steam used for purposes other than the operation of the plant under test. Losses of heat due to imperfections of the plant should be charged to the plant, and only such losses as are concerned in the working of the engine alone should be charged to the engine.

In measuring jacket water or any supply under pressure which has a temperature exceeding 212° F., the water should first be cooled, as may be done by discharging it into a tank of cold water previously weighed, or by passing it through a coil of pipe submerged in running and colder water, preventing thereby the loss of evaporation which occurs when such hot water is discharged into the open air.

XI. MEASUREMENT OF STEAM USED BY AUXILIARIES. — Although the steam used by the auxiliaries — embracing the air pump, circulating pump, feed pump, and any other apparatus of this nature, supposing them to be steam-driven, also the steam jackets, reheaters, etc., which consume steam required for the operation of the engine — is all included in the measurement of the steam consumption, as pointed out in Article X, yet it is highly desirable that the quantity of steam used by the auxiliaries, and in many cases that used by each auxiliary, should be determined exactly, so that the net consumption of the main engine cylinders may be ascertained and a complete analysis made of the entire work of the engine plant. Where the auxiliary cylinders are non-condensing, the steam consumption can often be measured by carrying the exhaust for the purpose into a tank of cold water resting on scales or through a coil of pipe surrounded by cold running water. Another method is to run the auxiliaries as a whole, or one by one, from a spare boiler (preferably a small vertical one), and measure the feed water supplied to this boiler. The steam used by the air and circulating pumps may be measured by running them under, as near as possible, the working conditions and speed, the main engine and other auxiliaries being stopped, and testing the consumption by the measuring apparatus used on the main trial. For a short trial, to obtain approximate results, measurement can be made by the water-gauge glass method, the feed supply being shut off. When the engine has a surface condenser, the quantity of steam used by the auxiliaries may be ascertained by allowing the engine alone to exhaust into the condenser, measuring the feed water supplied to the boiler and the water discharged by the air pump, and subtracting one from the other, after allowing for losses by leakage.

XII. COAL MEASUREMENT. — (*a*) *Commercial Tests.* — In commercial tests of the combined engine and boiler equipment, or those made under ordinary conditions of commercial service, the test should, as pointed out in Article VII, extend over the entire period of the day; that is, twenty-four hours, or a number of days of that duration. Consequently, the coal consumption should be determined for the entire time. If the engine runs but a part of the time, and during the remaining portion the fires are banked, the measurement of coal should include that used for banking. It is well, however, in such cases, to determine separately the amount consumed during the time the engine is in operation and that consumed during the period the fires are banked, so as to have complete data for purposes of analysis and comparison, using suitable precautions to obtain reliable measurements. The measurement of coal begins with the first firing, after cleaning the furnaces and burning down at the beginning of the test, as pointed out in Article VIII, and ends with the last firing, at the expiration of the allotted time.

(*b*) *Continuous Running Tests.* — In continuous running tests which, as pointed out in Article VII, cover one or more periods which elapse between the cleaning of the fires, the same principle applies as that mentioned under the above heading (*a*); viz., the coal measurement begins with the first firing, after cleaning and burning down, and the measurement ends with the last firing, before cleaning and burning down at the close of the trial.

(*c*) *Coal Tests in General.* — When not otherwise specially understood, a coal test of a combined engine and boiler plant is held to refer to the commercial test above noted, and the measurement of coal should conform thereto.

In connection with coal measurements, whatever the class of tests, it is important to ascertain the percentage of moisture in the coal, the weight of ashes and refuse, and, where possible, the approximate and ultimate analysis of the coal, following all the methods and details advocated in the latest report of the Boiler Test Committee of the Society. See also Chap. XIII of this book.

(*d*) *Other Fuels than Coal.* — For all other solid fuels than coal the same directions in regard to measurement should be followed as those given for coal If the boilers are run with oil or gas, the measurements relating to stopping and starting are much simplified, because the fuel is burned as fast as supplied, and there is no body of fuel constantly in the furnace, as in the case of using solid fuel. When oil is used, it should be weighed, and when gas is used, it should be measured in a calibrated gas meter or a gasometer.

XIII. INDICATED HORSE POWER. — The indicated horse power should be determined from the average mean effective pressure of diagrams taken at intervals of twenty minutes, and at more frequent intervals if the nature of the test makes this necessary, for each end of each cylinder.* With variable loads, such as those of engines driving generators for electric railroad work, and of rubber-grinding and rolling-mill engines, the diagrams cannot be taken too often. In cases like the latter, one method of obtaining suitable averages is to take a series of diagrams on the same blank card without unhooking the driving cord, and apply the pencil at successive intervals of ten seconds until two minutes' time or more has elapsed, thereby obtaining a dozen or more indications in the time covered. This tends to insure the determination of a fair average for that period. In taking diagrams for variable loads, as indeed for any load, the pencil should be applied long enough to cover several successive revolutions, so that the variations produced by the action of the governor may be properly recorded. To determine whether the governor is subject to what is called "racing" or "hunting," a "variation diagram" should be obtained; that is, one in which the pencil is applied a sufficient time to cover a complete cycle of variations. When the governor is found to be working in this manner, the defect should be remedied before proceeding with the test.

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* AUTHOR'S NOTE. This method of computing I.H.P. gives an acceptable result only when the speed of the engine is fairly constant. In case the speed fluctuates, not the average of the M.E.P. determinations should be used, but the average of the *pn* products in the equation

$$\text{I.H.P.} = \frac{plan}{33{,}000},$$

in which p = M.E.P. and n = revolutions per min.

It is seldom necessary, as far as average power measurements are concerned, to obtain diagrams at precisely the same instant at the two ends of the cylinder, or at the same instant on all the cylinders, when there are more than one. All that is required is to take the diagrams at regular intervals. Should the diagrams vary so much among themselves that the average may not be a fair one, it signifies that they should be taken more frequently, and not that special care should be employed to obtain the diagrams of each set at precisely the same time. When diagrams are taken during the time when the engine is working up to speed at the start, or when a study of valve setting and steam distribution is being made, they should be taken at as nearly the same time as practicable. In cases where the diagrams are to be taken simultaneously, the best plan is to have an operator stationed at each indicator. This is desirable, even where an electric or other device is employed to operate all the instruments at once; for unless there are enough operators, it is necessary to open the indicator cocks some time before taking the diagrams and run the risk of clogging the pistons and heating the high-pressure springs above the ordinary working temperature.

The most satisfactory driving rig for indicating seems to be some form of well-made pantograph, with driving cord of fine annealed wire leading to the indicator.* The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal lengths when the same indicator is attached to either end of the cylinder, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

With a perfect working pantograph, or similar apparatus, the equality in the length of diagrams taken with the same indicator at the two ends is sufficient indication of the substantial reliability of the reduction when the point of cut-off on the diagram is not unusually short — say, not shorter than one-eighth. When the cut-off is unusually short, the error produced by imperfect reduction becomes a comparatively large item, and one which for accurate work should be allowed for To test the accuracy of the reducing motion without making special preparations for a thorough examination, it is sufficient to make a comparison between the actual proportion of the stroke covered and the apparent proportion measured on the indicator, and see how they agree. This may be done on a large engine by making the comparison wherever it happens to stop, and repeating the comparison when it has stopped with the piston at some other point of the stroke. With an engine which can be turned over by hand, or where auxiliary power is provided for moving it, the comparison may be made at a number of equidistant points in the stroke. To make the test properly, a diagram should be taken just before stopping, and this will serve as a reference for the measurements taken after stopping. The actual proportion of stroke covered is determined by measuring the distance which the crosshead has moved and comparing it with the whole length of the stroke, making sure that the slack has all been taken up by turning sufficient steam into the cylinder to bring a pressure to bear on the piston, but not sufficient to start the flywheel in motion. To obtain the apparent indication from the diagram, the indicator pencil is moved up and down with the finger so as to make a vertical mark on the diagram, and the distance of this mark from the beginning of the diagram compared to the whole length of the diagram is the proportion desired.

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* For different types of reducing motions see Chap. XV of this book.

It is necessary, of course, to go through these operations without changing in any way the adjustment of the driving cord of the indicator, or any part of the mechanism that would alter the movements of the indicator.

The use of a three-way cock and a single indicator connected to the two ends of the cylinder is not advised, except in cases where it is impracticable to use an indicator close to each end. If a three-way cock is used, the error produced should be determined and allowed for.

The effect of the error produced by a three-way cock is usually to increase the area of the diagram. This is due to the tardiness of the indicator in responding to the changes of pressure. In an investigation made by one of the Committee, which was carried out both on short-stroke engines running at high speed and long-stroke engines running at comparatively slow speed, it was found that the increased area of the diagram, due to the sluggish action produced by the three-way cock, ranged from 3 to 7 per cent as compared with an indicator with a short and direct pipe.

In the manipulation of the indicator it is important to keep the instrument in clean condition and preserve it in mechanically good order. Ordinary cylinder oil is the best material to use for lubricating the indicator piston for pressures above the atmosphere. It is better to have the piston fit the cylinder rather loosely — so as to get absolute freedom of motion — than to have a mathematically accurate fit. In the latter case, extreme care and frequent cleanings are required to obtain good diagrams. No diagrams should be accepted in which there is any appearance of want of freedom in the movement of the mechanism. A ragged or serrated line in the region of the expansion or compression lines is a sure indication that the piston or some part of the mechanism sticks; and when this state of things is revealed the indicator should not be trusted, but the cause should be ascertained and a suitable remedy applied. Entire absence of wire drawing of the steam line, and especially a sharp, square corner at the beginning of the steam line, should be looked upon with suspicion, however desirable and satisfactory these features might otherwise be. These are frequently produced by an indicator which is defective, owing to want of freedom in the mechanism. An indicator which is free when subjected to a steady steam pressure, as it is under a test of the springs for calibration, should be able to produce the same horizontal line, or substantially the same, after pushing the pencil down with the finger, as that traced after pushing the pencil up and subsequently tapping it lightly. When the pencil is moved by the finger, first up and then down, the piston being subjected to pressure, the movement should appear smooth to the sense of feeling.

The point selected for attaching an indicator to the cylinder should never be the drip pipe or any point where the water of condensation will run into the instrument, if this can possibly be avoided. The admission of water with the steam may greatly distort the diagram. If it becomes necessary to place the indicator in such a position, as may happen when it is attached to the lower end of a vertical cylinder, the connection to the indicator must be short and direct, and in some cases it should be provided with a drip chamber arranged so as to collect the water or deflect it from entering the instrument.

In all cases the pipes leading to indicators should be as short and direct as possible.

To determine the average power developed in cases where the engine starts from rest during the progress of the trial, as in a commercial test of a plant where the engine runs only a portion of the twenty-four hours, a number of diagrams should be taken during the period of getting up speed and applying the working load, the corresponding speed for each set of diagrams being counted. The power shown by these diagrams for the proportionate time should be included in the average for the whole run, and the duration should be the time the throttle valve is open.

XIV. TESTING INDICATOR SPRINGS. — See Arts. 329, 330, p. 593, of this book, which practically cover the provisions of the Code.

XV. BRAKE HORSE POWER. — See Chap. X of this book, for construction of brakes, computation of power, etc.

XVI. QUALITY OF STEAM. — The calorimeter, of whatever type, should be attached to the main steam pipe close to the throttle. Concerning the different types of calorimeters and the method of using them, see Chap. XIV of this book.

The Code recommends that the calorimeter be attached to a vertical pipe if possible, in which case the perforated sampling tube, Fig. 374, may be used. Where it is necessary to attach to a horizontal pipe, it is further recommended that the sampling tube be inserted through the bottom of the pipe through a stuffing box, and that readings be taken at different points in the cross section of the steam pipe, the sampling tube being open of course only at its end. If the reading near the bottom of the pipe should show water, it is recommended that a drip pipe be inserted just ahead of the calorimeter, through which pipe a sufficient quantity of steam is continuously removed to take out the water. Due allowance of course should be made for this. If this scheme does not result in fairly dry steam being shown in the calorimeter at all positions of the sampling tube, or if in the vertical pipe the indications vary greatly, sometimes exceeding 3 per cent, it is then recommended to insert a separator.

XVII. SPEED. — There are several reliable methods of ascertaining the speed, or the number of revolutions of the engine crank-shaft per minute. The simplest is the familiar method of counting the number of turns for a period of one minute with the eye fixed on the second hand of a timepiece. Another is the use of a counter held for a minute or a number of minutes against the end of the main shaft. Another is the use of a reliable calibrated tachometer held likewise against the end of the shaft. The most reliable method, and the one we recommend, is the use of a continuous recording engine register or counter, taking the total reading each time that the general test data are recorded, and computing the revolutions per minute corresponding to the difference in the readings of the instrument. When the speed is above 250 revolutions per minute, it is almost impossible to make a satisfactory counting of the revolutions without the use of some form of mechanical counter.

The determination of variation of speed during a single revolution, or the effect of the fluctuation due to sudden changes of the load, is also desirable, especially in engines driving electric generators used for lighting purposes. There is at present no recognized standard method of making such determinations, and if such are desired, the method employed may be devised by the person making the test, and described in detail in the report.

One method suggested for determining the instantaneous variation of speed which accompanies a change of load is as follows: A screen containing a narrow slot is placed on the end of a bar and vibrated by means of electricity. A corresponding slot in a stationary screen is placed parallel and nearly touching the vibrating screen, and the two screens are placed a short distance from the flywheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel. The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots. When this is done the observer viewing the wheel through the slots sees what appears to be a stationary flywheel. When a change in the velocity of the flywheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change. By careful observations of the amount of this motion, the angular change of velocity during any given time is revealed.

Experiments that have been made with a device of this kind show that the instantaneous gain of velocity, upon suddenly removing all the load from an engine, amounted to from one-sixth to one-quarter of a revolution of the wheel.*

XVIII. Recording the Data. — Take note of every event connected with the progress of the trial, whether it seems at the time to be important or unimportant. Record the time of every event, the time of taking every weight, and every observation. Observe the pressures, temperatures, water heights, speeds, etc., every twenty or thirty minutes when the conditions are practically uniform, and at much more frequent intervals if the conditions vary. Observations which concern the feed-water measurement should be made with special care at the expiration of each hour of the trial, so as to divide the tests into hourly periods and show the uniformity of the conditions and results as the test goes forward. Where the water discharged from a surface condenser is weighed it may be advisable to divide the test by this means into periods of less than one hour.

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* For further information concerning the Measurement of Speed, see Chap. VIII of this book. Attempts to study governor action under changing loads have sometimes been made by use of the ordinary indicating tachometer. No reliance can be placed on such indications, and it usually becomes necessary to arrange for a time indication by seconds pendulum, or other means, together with an automatic recording of the revolutions.

The data and observations of the test should be kept on properly prepared blanks or in notebooks containing columns suitably arranged for a clear record. As different observers have their own individual ideas as to how such records should be kept, no special form of log sheet is given as a necessary part of the code.*

XIX. UNIFORMITY OF CONDITIONS. — In a test, having for an object the determination of the maximum economy obtainable from an engine, or where it is desired to ascertain with special accuracy the effect of predetermined conditions of operation, it is important that all the conditions under which the engine is operated should be maintained uniformly constant. This requirement applies especially to the pressure, the speed, the load, the rate of feeding the various supplies of water, the height of water in the gauge glasses, and the depth of water in the feed-water reservoir.

XX. ANALYSIS OF THE INDICATOR DIAGRAMS. — (*a*) *Steam Accounted for by the Indicator.*—This is also known as the "Diagram Water Rate." See Art. 348, p. 653, of this book.

(*b*) *Sample Indicator Diagrams.*— In order that the report of a test may afford complete information regarding the conditions of the test, sample indicator diagrams should be selected from those taken and copies appended to the tables of results. In cases where the engine is of the multiple-expansion type these sample diagrams may also be arranged in the form of a "combined" diagram.

The Code next shows methods of obtaining combined diagrams. See Art. 353, p. 661, of this book.

(*c*) *The Point of Cut-off.*— The term "cut-off" as applied to steam engines, although somewhat indefinite, is usually considered to be at an earlier point in the stroke than the beginning of the real expansion line. That the cut-off point may be defined in exact terms for commercial purposes, as used in steam-engine specifications and contracts, the Committee recommends that, unless otherwise specified, the *commercial cut-off*, which seems to be an appropriate expression for this term, be ascertained as follows: Through a point showing the maximum pressure during admission, draw a line parallel to the atmospheric line. Through the point on the expansion line near the actual cut-off, draw a hyperbolic curve. The point where these two lines intersect is to be considered the *commercial cut-off* point. The percentage is then found by dividing the length of the diagram measured to this point, by the total length of the diagram, and multiplying the result by 100.

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* See the log forms used in Sibley College, pp. 771, 772, and 773 of this book.

The principle involved in locating the commercial cut-off is shown in Figs. 494 and 495, the first of which represents a diagram from a slow-speed Corliss engine, and the second a diagram from a single-valve high-speed engine. In the latter case, where, owing to the fling of the pencil, the steam line vibrates, the maximum pressure is found by taking a mean of the vibrations at the highest point.

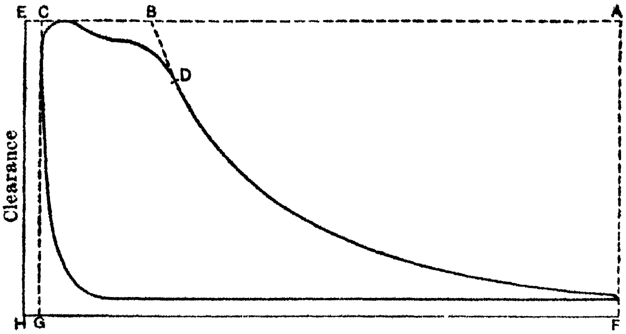

FIG. 494.

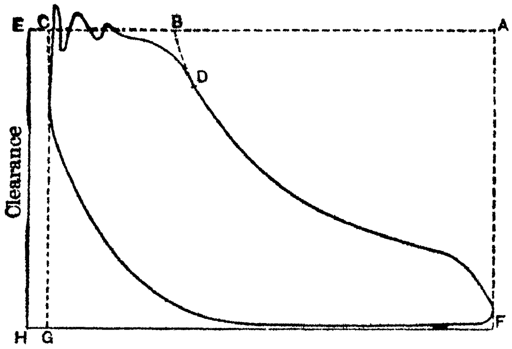

FIG. 495.

The *commercial cut-off*, as thus determined, is situated at an earlier point of the stroke than the actual cut-off referred to in computing the "steam accounted for" by the indicator in Section XX (*a*).

(*d*) *Ratio of Expansion.*—The ratio of expansion for a simple engine is determined by dividing the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple-expansion engine it is determined by dividing the net volume of the steam indicated by the low-pressure diagram at the end of the expansion line, assumed to be continued to the end of the stroke, by the net volume of the steam at the maximum pressure during admission to the high-pressure cylinder.

For example, in a simple engine, referring to Figs. 494 and 495, the ratio of expansion is the entire distance *HF* including clearance, divided by the distance *EB* including clearance; that is, $\frac{HF}{EB}$.

For a compound engine, referring to the combined diagram, Fig. 496, the ratio of expansion is the distance *CD* divided by the distance *AB*; in which

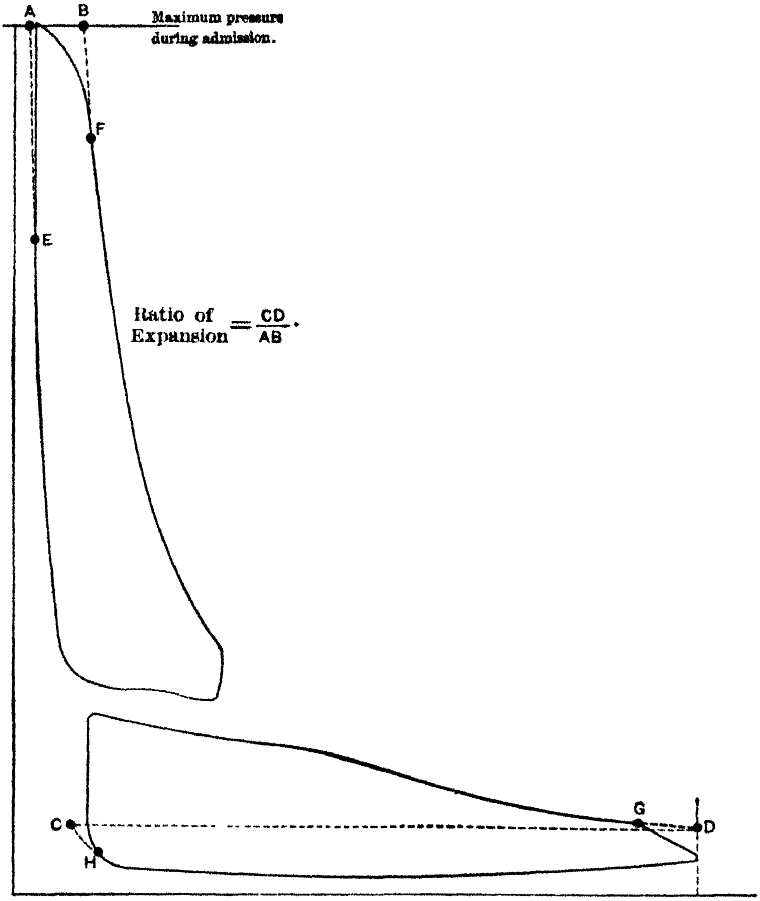
A
B
Maximum pressure during admission.
F
E
Ratio of Expansion = CD/AB.
G
C
D
H

FIG. 496.

E and *F* are points on the compression and expansion lines respectively of the high-pressure diagram, the latter being near the point of cut-off; and *H* and *G*, points on the compression and expansion lines of the low-pressure diagram, the latter being near the point of release, and the curves *EA*, *FB*, *HC*, and *GD* being hyperbolic. If it is desired to determine the ratio without laying out the combined diagram it can be done by drawing on the original diagrams the hyperbolic curves referred to above, and multiplying the ratio of volumes of the cylinders, first by the ratio of the length of the high-pressure diagram to the distance *AB* and then by the ratio of the distance *CD* to the length of the low-pressure diagram.

(*e*) *Diagram Factor.* — The diagram factor is the proportion borne by the actual mean effective pressure measured from the indicator diagram to that of a diagram in which the various operations of admission, expansion, release, and compression are carried on under assumed conditions. The factor recommended refers to an ideal diagram which represents the maximum power obtainable from the steam accounted for by the indicator diagrams at the point of cut-off, assuming first that the engine has no clearance; second, that there are no losses through wire-drawing the steam either during the admission or the release; third, that the expansion line is a hyperbolic curve; and fourth, that the initial pressure is that of the boiler and the back pressure that of the atmosphere for a non-condensing engine, and of the condenser for a condensing engine.

The diagram factor is useful for comparing the steam distribution losses in different engines, and is of special use to the engine designer, for by multiplying the mean effective pressure obtained from the assumed theoretical diagrams by it he will obtain the actual mean effective pressure that should be developed in an engine of the type considered. The expansion and compression curves are taken as hyperbolas, because such curves are ordinarily used by engine builders in their work, and a diagram based on such curves will be more useful to them than one where the curves are constructed according to a more exact law.

In cases where there is a considerable loss of pressure between the boiler and the engine, as where steam is transmitted from a central plant to a number of consumers, the pressure of the steam in the supply main should be used in place of the boiler pressure in constructing the diagrams.

The method of determining the diagram factor is best shown by referring to Figs. 497 to 500, which apply to a simple non-condensing engine, a simple condensing engine, and a compound condensing engine.

In Fig. 497 *RS* represents the volume of steam at boiler pressure admitted to the cylinder, *PR* and *OS* being hyperbolic curves drawn through the compression and cut-off points respectively. In Fig. 498 the factor is the proportion borne by the area of the actual diagram to that of the diagram *CNHSK*. In Fig. 499 the factor is the proportion borne to the area of the diagram *CNHSK*. In Fig. 500 the factor is the proportion borne by the area of the two combined diagrams to the area *CNHSK*. In Fig. 498, where the diagram is the same as in Fig. 497, the distance *CN* is laid off equal to *RS* shown in Fig. 497, and the curve *NH* is a hyperbola referred to the zero lines *CM* and *MJ*. In Fig. 499

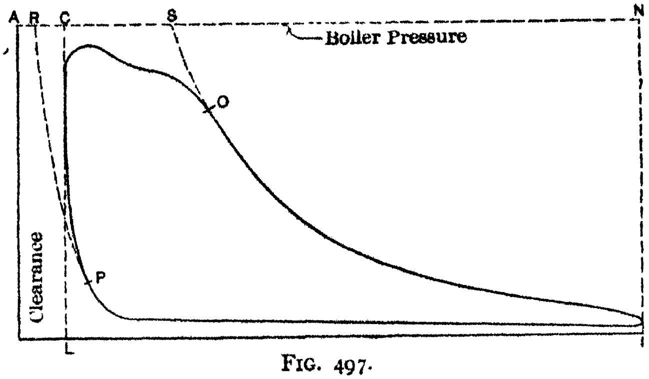

FIG. 497.

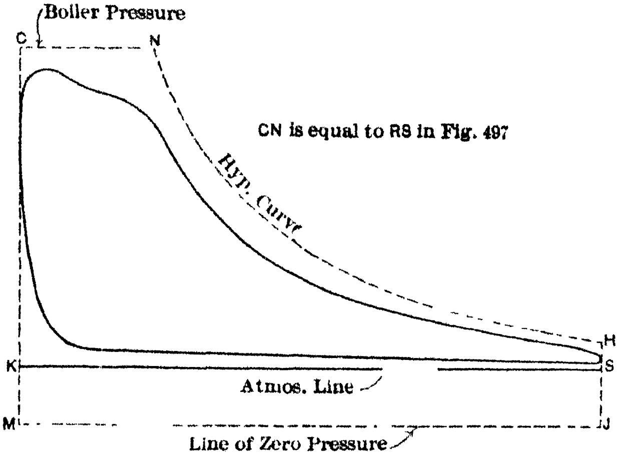

FIG. 498.

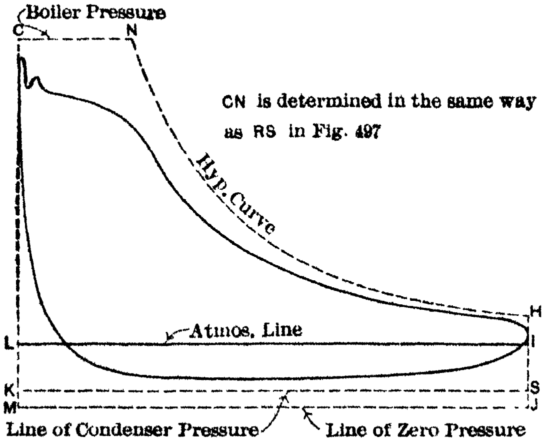
Boiler Pressure
C
N
CN is determined in the same way as RS in Fig. 497
Hyp. Curve
H
Atmos. Line
L
I
K
S
M
J
Line of Condenser Pressure
Line of Zero Pressure

FIG. 499.

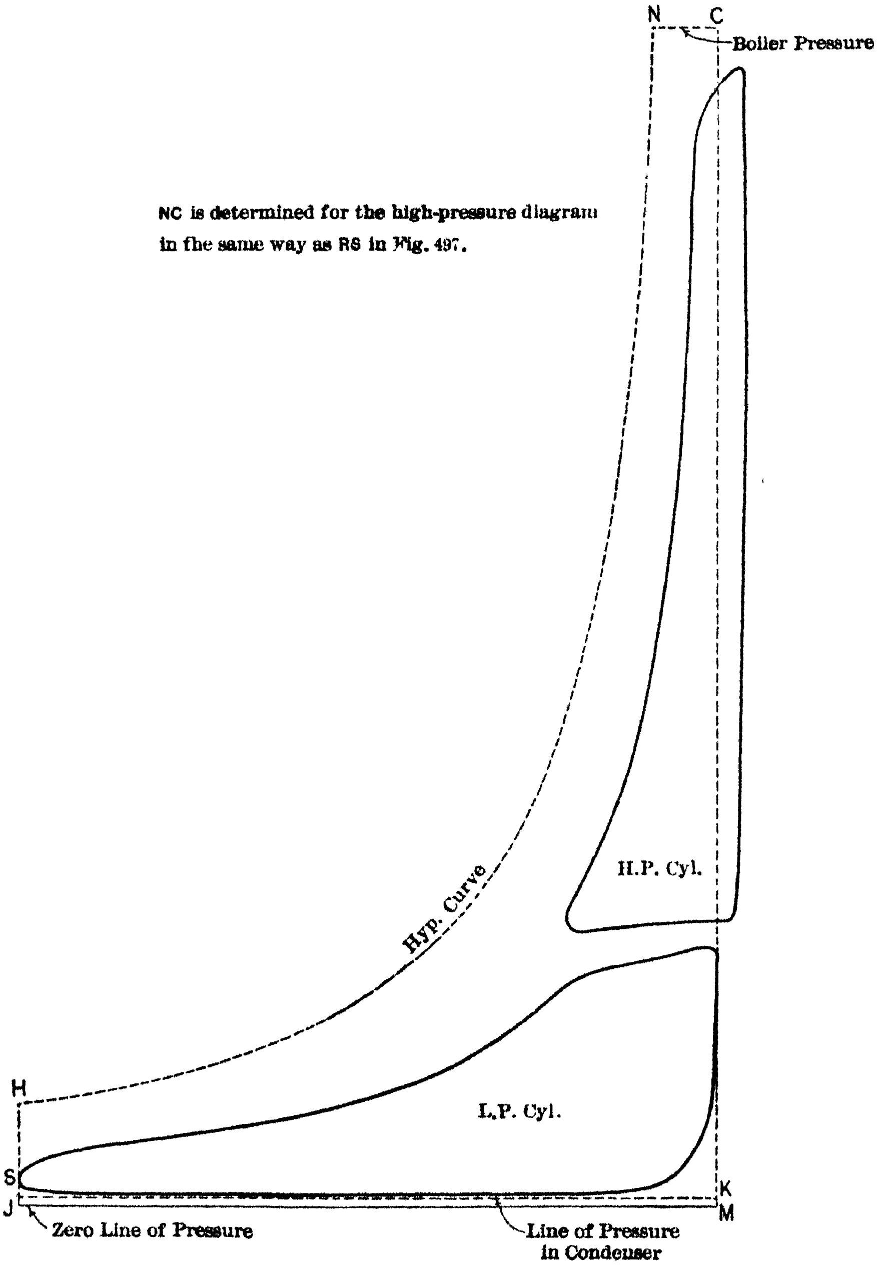
N
C
Boiler Pressure
NC is determined for the high-pressure diagram in the same way as RS in Fig. 497.
Hyp. Curve
H.P. Cyl.
L.P. Cyl.
H
S
J
K
M
Zero Line of Pressure
Line of Pressure in Condenser

FIG. 500.

the distance CN is found in a similar way. In Fig. 500 the distance CN for the high-pressure cylinder is found in the same manner as in the case of a simple engine. The mean effective pressure of the ideal diagram can readily be obtained from the formula

$$\frac{P}{R}(1 + \text{Hyp. Log. R.}) - p. \qquad (1)$$

where P is the absolute pressure of the steam in the boiler, R the ratio $\frac{MJ}{CN}$, and p the pressure of the atmosphere or in the condenser.

XXI. STANDARDS OF ECONOMY AND EFFICIENCY. — The hourly consumption of heat, determined by employing the actual temperature of the feed water to the boiler, as pointed out in Article IX of the Code, divided by the indicated and brake horse power, that is, the number of heat units consumed per indicated and per brake horse power per hour, are the standards of engine efficiency recommended by the Committee. The consumption *per hour* is chosen rather than the consumption per minute, so as to conform with the designation of time applied to the more familiar units of coal and water measurement, which have heretofore been used. The British standard, where the temperature of the feed water is taken as that corresponding to the temperature of the back-pressure steam, allowance being made for any drips from jackets or reheaters, is also included in the tables.

It is useful in this connection to express the efficiency in its more scientific form, or what is called the "thermal efficiency ratio." The thermal efficiency ratio is the proportion which the heat equivalent of the power developed bears to the total amount of heat actually consumed, as determined by test. The heat converted into work represented by one horse power is 1,980,000 foot-pounds per hour, and this divided by 778 equals 2545 B.t.u. Consequently, the thermal efficiency ratio is expressed by the fraction

$$\frac{2545}{\text{B.t.u. per H P. per hour}}.$$

MEMORANDA REGARDING THE BRITISH STANDARD. — The principal objection which the Committee has to the use of the British standard of engine economy for the leading place is as follows:

The practical utility of an engine depends almost wholly upon the fact that it is used for some form of industrial work, and, in combination with a boiler and various appurtenances, it is a part of a complete power plant. Were it not for the unreliable character of coal and other fuels, the proper standard of economy for such an engine, with a boiler of given efficiency, would be the fuel consumed, because fuel is, in reality, the source of the power, and this is the most important thing to be supplied in operating the engine. The use of a standard of heat units in place of fuel meets the objectionable characteristics of coal due to its heat variability, and in doing this it makes no change in the conditions under which the standard should be employed. These are the actual conditions of use, and not the ideal conditions. The British method furnishes a means of determining and stating the ideal performance of an engine working under assumed conditions. It does not give the performance of the entire engine plant under the actual conditions as affected by the efficiency of the heaters and auxiliaries, and the performance under such conditions is usually of the greatest importance.

The advantages in favor of the British standard are as follows:

1. It expresses the economy of the engine as an independent machine, unaffected by the failings of the feed-water heater or condenser.

2. In this form it is useful for expressing the economy of an engine in the testing shop of an engine builder, or in the mechanical laboratory of a college, where it forms no part of a power plant.

3. A comparison of the British standard with the economy of an ideal engine working between the same limits of temperature reveals those losses going on which are due to the engine alone, and this furnishes information not otherwise obtained.

The arrangement of the various forms used in stating the efficiency, as proposed by the Committee, gives due weight to both of these standards, placing the one in the lead which is regarded as the most important.

XXII. HEAT ANALYSIS. — For certain scientific investigations it is useful to make a heat analysis of the diagram, to show the interchange of heat from steam to cylinder walls, etc., which is going on within the cylinder. This is unnecessary for commercial tests.

For details of the method of carrying out such heat analysis, see Art. 365, p. 799 of this book.

XXIII. TEMPERATURE-ENTROPY DIAGRAM. — For method of drawing such a diagram and explanation of what it shows, see Art. 354, p. 670.

XXIV. RATIO OF ECONOMY OF AN ENGINE TO THAT OF AN IDEAL ENGINE. — The ideal engine recommended for obtaining this ratio is that which was adopted by the Committee appointed by the Civil Engineers of London to consider and report a standard thermal efficiency for steam engines. This engine is one which follows the Rankine cycle, where steam at a constant pressure is admitted into the cylinder with no clearance, and, after the point of cut-off, is expanded adiabatically to the back pressure. In obtaining the economy of this engine the feed water is assumed to be returned to the boiler at the exhaust temperature. Such a cycle is preferable to the Carnot for the purpose at hand, because the Carnot cycle is theoretically impossible for an engine using superheated steam produced at a constant pressure, and the gain in efficiency for superheated steam corresponding to the Carnot efficiency will be much greater than that possible for the actual cycle.

The ratio of the economy of an engine to that of the ideal engine is obtained by dividing the heat consumption per indicated horse power per minute for the ideal engine by that of the actual engine.*

XXV. Miscellaneous. — In the case of tests of combined engines and boiler plants, where the full data of the boiler performance is to be determined, reference should be made to the directions given by the Boiler Test Committee of the Society, Code of 1899. (See Chap. XVII of this book.)

In tests made for scientific research, and in those made on special forms of engines, the line of procedure must be varied according to the special objects in view, and it has been deemed unnecessary to go into particulars applying to such tests

In testing steam pumping engines and locomotives in accordance with the standard methods of conducting such tests, recommended by the committees of the Society, reference should be made to the reports of those committees in the "Transactions." (See also pages following).

XXVI. Report of Test. — The data and results of the test should be reported in the manner and in the order outlined in one of the following tables, the first of which gives, it is hoped, a complete summary of all the data and results as applied not only to the standard heat-unit test, but also to tests of combined engine and boiler for determining all questions of performance, whatever the class of service; the second refers to a short form of report giving the necessary data and results for the standard heat test; and the third to a short form of report for a feed-water test. It is the intention that the tables should be full enough to apply to any type of engine, but where not so, or where special data and results are determined, additional results may be inserted under the appropriate headings. Although these forms are arranged so as to be used for expressing the principal data and results of tests of pumping engines and locomotives, as well as for all other classes of steam engines, it is not the intention that they shall supplant the forms recommended by the committees on Duty Trials and Locomotives, in cases where the full report of a test of such engines is desired.

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* The Code gives means of computing the heat consumption of the ideal cycle described, reproducing a set of curves from the Minutes of the Proceedings of the Civil Engineers of London. It is not thought necessary to reproduce this here, especially since the Mollier Diagram, Fig. 245, gives practically the same information, based upon later data.

To show the method of using the Mollier chart, take the following examples:

Steam pressure 135 lbs. abs.; back pressure 15 lbs. abs. Assume steam dry and saturated at cut-off. The chart shows that the heat content of the steam at the higher pressure is 1193 B.t.u. After adiabatic (isentropic) expansion to 15 lbs., the heat content is 1031 B.t u. Out of each pound of steam, 1193 − 1031 = 162 B.t.u. are therefore converted into work. The number of pounds of steam required per I.H.P. hour, when operating on the ideal cycle, will then be $\frac{2545}{162}$ = 15.71 lbs., and the heat consumption of the ideal engine per I.H.P. hour will then be 1193 × 15.71 = 18,742 B.t.u.

If the steam at 135 lbs. abs. pressure had been superheated to 500° (= 150° of superheat), and expanded to 15 lbs. abs., as before, the figures would have been: Heat content at 135 lbs. and 500° = 1273 B.t.u.; heat content at 15 lbs. abs. = 1093 B.t.u.; heat converted into work per pound of steam = 180 B.t.u.; steam consumption of ideal engine = $\frac{2545}{180}$ = 14.14 pounds; heat consumption of ideal engine = 14.14 × 1273 = 18,000 B.t.u. per hour.

The ideal cycle above outlined is by many authorities called the *Clausius* instead of the *Rankine* cycle. The latter is then defined as the cycle in which isentropic expansion is not carried to back pressure as in the Clausius, but stops at some terminal pressure higher than the back pressure. See the discussion on theoretical cycles, in Chap. XI of this book.

It is recommended that any report be supplemented by a chart in which the data of the test is graphically presented.

TABLE NO. I.

Data and Results of Steam-Engine Test.

Arranged according to the Complete Form advised by the Engine Test Committee of the American Society of Mechanical Engineers. Code of 1902.

1. Made by .... of ....
   on engine located at ....
   to determine ....
   ....
2. Date of trial ....
3. Type of engine (simple, compound, or other multiple expansion; condensing or non-condensing) ....
   ....
4. Class of engine (mill, marine, locomotive, pumping, electric, or other) ..
5. Rated power of engine ....
6. Name of builders ....
7. Number and arrangement of cylinders of engine; how lagged; type of condenser ....
8. Type of valves ....
9. Type of boiler ....
10. Kind and type of auxiliaries (air, circulating, main, and feed pumps; jackets, heaters, etc.) ....
    ....
    ....

	1st Cyl.	2d Cyl	3d Cyl
11. Dimensions of engine			
(a) Single or double acting			
(b) Cylinder dimensions:			
Bore .... in.			
Stroke .... ft.			
Diameter of piston rod .... in.			
Diameter of tail rod .... in.			

	1st Cyl.	2d Cyl.	3d Cyl.
(*c*) Clearance in per cent of volume displaced by piston per stroke:			
Head end...........................			
Crank end..........................			
Average..........................			
(*d*) Surface in square feet (average).........			
Barrel of cylinder....................			
Cylinder heads			
Clearance and ports......			
Ends of piston....			
(*e*) Jacket surfaces or internal surfaces of cylinder heated by jackets, in square feet:			
Barrel of cylinder.......			
Cylinder heads			
Clearance and ports.			
Receiver jackets			
(*f*) Ratio of volume of each cylinder to volume of high-pressure cylinder.			
(*g*) Horse-power constant for one pound mean effective pressure, and one revolution per minute			

12. Dimensions of boilers:
(*a*) Number.............. ....
(*b*) Total grate surface .... q. ft
(*c*) Total water-heating surface (external) .... "
(*d*) Total steam-heating surface (external) .... "

13. Dimension of auxiliaries:
(*a*) Air pump.. ....
....
(*b*) Circulating pump ....
....
(*c*) Feed pumps...
....
(*d*) Heaters..
....

14. Dimensions of condenser...
....

15. Size, length, and number of turns in main steam pipe leading from the boiler to the engine...
....

16. Give description of main features of plant, and illustrate with drawings to be given on an appended sheet.

Total Quantities, Time, Etc.

17. Duration of test............................ hours.
18. Length of time engine was in motion with throttle open.. "
19. Length of time engine was running at normal speed.... "
20. Water fed to boilers from main source of supply..... lbs.
21. Water fed from auxiliary supplies:
    (*a*) .... "
    (*b*) .... "
    (*c*) .... "
22. Total water fed to boilers from all sources .... "
23. Moisture in steam or superheating near throttle .....per cent or deg.
24. Factor of correction for quality of steam, dry steam being unity..
25. Total dry steam consumed for all purposes * ...... lbs.
26. Total coal as fired † .... "
27. Moisture in coal............................. per cent.
28. Total dry coal consumed....................... lbs
29. Ash and refuse .... "
30. Percentage of ash and refuse to dry coal........ per cent.
31. Calorific value of coal by calorimeter test per pound of dry coal, determined by ... . calorimeter . . B.t.u.
32. Cost of coal per ton of 2240 lbs. .... $

Hourly Quantities.

33. Water fed from main source of supply .... lbs.
34. Water fed from auxiliary supplies:
    (*a*) .... "
    (*b*) .... "
    (*c*) .... "
35. Total water fed to boilers per hour. .... "
36. Total dry steam consumed per hour .... "
37. Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant............ "
38. Net dry steam consumed per hour by engine and auxiliaries "
39. Dry steam consumed per hour:
    (*a*) Main cylinders............................ "
    (*b*) Jackets and reheaters...................... "
    (*c*) Air pump .... "

---

* In case of superheated steam engines, determine, if practicable, the temperature of the steam in each cylinder.

† Where an independent superheater is used, this includes coal burned in the superheater.

(*d*) Circulating pump........ lbs.
(*e*) Feed-water pump........ "
(*f*) Other auxiliaries........ "
40. Dry coal consumed per hour:
(*a*) During running period........ "
(*b*) During banking period........ "
(*c*) Total........ "
41. Injection or circulating water supplied condenser per hour. cu. ft.

Pressures and Temperatures (Corrected).

42. Steam pressure at boiler by gauge........ lbs. per sq. in.
43. Steam-pipe pressure near throttle, by gauge........ "
44. Barometric pressure of atmosphere in inches of mercury ins.
45. Pressure in first receiver by gauge........ lbs. per sq. in.
46. Pressure in second receiver by gauge........ "
47. Vacuum in condenser:
(*a*) In inches of mercury........ ins.
(*b*) Corresponding total pressure........ lbs. per sq. in.
48. Pressure in steam jacket by gauge........ "
49. Pressure in reheater by gauge........ "
50. Moisture in steam or superheating at boilers........ p. c. or deg. F.
51. Superheating of steam in first receiver........ deg. Fahr.
52. Superheating of steam in second receiver........ "
53. Temperature of main supply of feed water to boilers........ "
54. Temperature of auxiliary supplies of feed water:
(*a*)........ "
(*b*)........ "
(*c*)........ "
55. Ideal feed-water temperature corresponding to the pressure of the steam in the exhaust pipe, allowance being made for heat derived from jacket or reheater drips........ "
56. Temperature of injection or circulating water entering condenser........ "
57. Temperature of injection or circulating water leaving condenser........ "
58. Temperature of chimney gases entering economizer........ "
59. Temperature of chimney gases leaving economizer........ "
60. Temperature of water entering economizer........ "
61. Temperature of water leaving economizer........ "
62. Temperature of air in boiler room........ "
63. Temperature of air in engine room........ "

Data Relating to Heat Measurements.

64. Heat units per pound of feed water, main supply........ B.t.u.
65. Heat units per pound of feed water, auxiliary supply:
    (*a*) .... “
    (*b*) .... “
    (*c*) .... “
66. Heat units consumed per hour, main supply........ “
67. Heat units consumed per hour, auxiliary supplies:
    (*a*) .... “
    (*b*) .... “
    (*c*) .... “
68. Total heat units consumed per hour for all purposes .... “
69. Loss of heat per hour due to leakage of plant, drips, etc . “
70. Heat units consumed per hour:
    (*a*) By engine alone .... “
    (*b*) By auxiliaries .... “
71. Heat units consumed per hour by the engine alone, reckoned from temperature given in line 55.............. “

Indicator Diagrams.

1st Cyl. 2d Cyl. 3d Cyl

72. Commercial cut-off in per cent of stroke.......
73. Initial pressure in lbs. per sq. in. above atmosphere.........................
74. Back-pressure at mid-stroke above or below atmosphere in lbs. per sq. in.. ....
75. Mean effective pressure in lbs. per sq. in........
76. Equivalent mean effective pressure in lbs. per sq. in.:
    (*a*) Referred to first cylinder...............
    (*b*) Referred to second cylinder.............
    (*c*) Referred to third cylinder..............
77. Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves...................
    Pressure above zero in lbs. per sq. in.:
    (*a*) Near cut-off..........................
    (*b*) Near release.........................
    (*c*) Near beginning of compression.........

1st. Cyl. 2d Cyl. 3d Cyl.

Percentage of stroke at points where pressures are measured:
(*a*) Near cut-off . . . . . .
(*b*) Near release . . . . . .
(*c*) Near beginning of compression . . . . . .
78. Aggregate M.E.P. in lbs. per sq. in. referred to each cylinder given in heading . . . . . .
79. Mean back-pressure above zero, lbs. per sq. in. . .
80. Steam accounted for in lbs. per indicated horse power per hour:
(*a*) Near cut-off . . . . . .
(*b*) Near release . . . . . .
81. Ratio of Expansion . . . . . .
82. Mean effective pressure of ideal diagram . . . . . . lbs. per sq. in.
83. Diagram factor . . . . . . per cent.

Speed.

84. Revolutions per minute . . . . . . rev.
85. Piston speed per minute . . . . . . ft.
86. Variation of speed between no load and full load . . . rev.
87. Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change . . . . . . "

Power.

88. Indicated horse power developed by main-engine cylinders:
First cylinder . . . . . . H.P.
Second cylinder . . . . . . "
Third cylinder . . . . . . "
Total . . . . . . "
89. Brake H.P., electric H.P., pump H.P., or dynamo H.P., according to the class of engine . . . "
90. Friction I.H.P. by diagrams, no load on engine, computed for average speed . . . . . . "
91. Difference between indicated H.P. and brake H.P. . . . . . . "
92. Percentage of indicated H.P. of main engine lost in friction. per cent.
93. Power developed by auxiliaries:
(*a*) . . . . . . H.P.
(*b*) . . . . . . "
(*c*) . . . . . . "

Standard Efficiency Results.

Heat units consumed by engine and auxiliaries per hour:
  (*a*) Per indicated horse power.................... B.t.u.
  (*b*) Per brake horse power.................... "

95. Equivalent standard coal consumed by engine and auxiliaries per hour, assuming calorific value such that 10,000 B.t.u. are imparted to the boiler per lb.:
  (*a*) Per indicated horse power.................... lbs.
  (*b*) Per brake horse power.................... "

96. Heat units consumed per minute:
  (*a*) Per indicated horse power.................... B.t.u.
  (*b*) Per brake horse power.................... "

97. Heat units consumed by engine per hour corresponding to ideal maximum temperature of feed water given in line 55, British standard:
  (*a*) Per indicated horse power.................... "
  (*b*) Per brake horse power.................... "

Efficiency Ratios.

98. Thermal efficiency ratio:
  (*a*) Per indicated horse power.................... per cent.
  (*b*) Per brake horse power.................... "
  (*c*) Ratio of efficiency of engine to that of an ideal engine working with the Rankine cycle.................... "

*Miscellaneous Efficiency Results.**

99. Dry steam consumed per indicated H.P. per hour:
  (*a*) Main cylinder, including jackets.................... lbs.
  (*b*) Auxiliary cylinders, etc.................... "
  (*c*) Engine and auxiliaries.................... "

100. Dry steam consumed per brake H.P. per hour:
  (*a*) Main cylinders, including jackets.................... lbs.
  (*b*) Auxiliary cylinders, etc.................... "
  (*c*) Engine and auxiliaries.................... "

101. Percentage of steam used by main-engine cylinders accounted for by indicator diagrams:

	1st Cyl.	2d Cyl.	3d Cyl.
(*a*) Near cut-off....................			
(*b*) Near release....................			

---

* The horse power on which the above efficiency results (items 94 to 103) are based is that of the main engine, exclusive of auxiliaries.

102. Dry coal consumed by combined engine and boiler plant per I.H.P. per hour:
  (*a*) During running period........................ lbs.
  (*b*) During banking period........................ "
  (*c*) Total.. "
103. Dry coal consumed by combined engine and boiler plant per brake H.P. per hour:
  (*a*) During running period........................ "
  (*b*) During banking period........................ "
  (*c*) Total.. "
104. Water evaporated under actual conditions per pound of dry coal........................ lbs.
105. Equivalent evaporation from and at 212° F. per pound of dry coal........................ "
106. Efficiency of boilers based on dry coal........................ per cent.
107. Combined efficiency of boiler and engine plant........................ "

Additional Calculations Recommended for Special Classes of Steam Engines.

Water-pumping Engines.

108. Duty per 1,000,000 heat units imparted to the boiler........ ft.-lbs.
109. Duty per 1000 pounds of dry steam........................ "
110. Duty per 100 pounds of actual coal consumed by plant........ "
111. Number of gallons of water pumped in 24 hours........ gals.

Locomotives.

112. Dynamometric horse power........................ H.P.
113. "Standard Coal" of 12,500 B.t.u. value consumed, per dynamometric H.P. per hour........................ lbs.

Electric-light Engines and those Driving
Generators for Electric Railways.

114. Current........................ amperes.
115. Electro-motive force........................ volts.
116. Electrical power generated in watts........................ watts.
117. Electrical horse power generated........................ H.P.
118. Efficiency of generator........................ per cent.
119. Heat units consumed per electrical horse power per hour B.t.u.
120. Dry steam consumed per electrical horse power per hour lbs.
121. Dry coal consumed per electrical horse power per hour:
  (*a*) During running period........................ lbs.
  (*b*) During banking period........................ "
  (*c*) Total.. "

Additional Data.

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is used. Also give copies of indicator diagrams nearest the mean and the corresponding scales.

TABLE NO. II.

DATA AND RESULTS OF STANDARD HEAT TEST OF STEAM ENGINE.

Arranged according to the Short Form advised by the Engine Test Committee of the American Society of Mechanical Engineers. Code of 1902.

1. Made by .... of ....
   on engine located at ....
   to determine ....
   ....
2. Date of trial ....
3. Type and class of engine; also of condenser ....
   ....

1st Cyl 2d Cyl. 3d Cyl.

4. Dimension of main engine:
   (*a*) Diameter of cylinder .... in.
   (*b*) Stroke of piston .... ft.
   (*c*) Diameter of piston rod .... in.
   (*d*) Average clearance .... p.c.
   (*e*) Ratio of volume of cylinder to high-pressure cylinder ....
   (*f*) Horse-power constant for one pound mean effective pressure and one revolution per minute ....
5. Dimensions and type of auxiliaries ....
   ....

Total Quantities, Time, Etc.

6. Duration of test .... hours.
7. Total water fed to boilers from main source of supply .... lbs.
8. Total water fed from auxiliary supplies:
   (*a*) .... "
   (*b*) .... "
   (*c*) .... "
9. Total water fed to boilers from all sources .... "
10. Moisture in steam or superheating near throttle .... p. c. or deg.
11. Factor of correction for quality of steam ....
12. Total dry steam consumed for all purposes .... lbs.

Hourly Quantities.

13. Water fed from main source of supply.................. lbs.
14. Water fed from auxiliary supplies:
    - (*a*) .... "
    - (*b*) .... "
    - (*c*) .... "
15. Total water fed to boilers per hour...................... "
16. Total dry steam consumed per hour........................ "
17. Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant.............. "
18. Net dry steam consumed per hour by engine and auxiliaries "

Pressures and Temperatures (Corrected).

19. Pressure in steam pipe near throttle by gauge .... lbs. per sq. in.
20. Barometric pressure of atmosphere in inches of mercury.. ins.
21. Pressure in receivers by gauge.......................... lbs. per sq. in.
22. Vacuum in condenser in inches of mercury................ ins.
23. Pressure in jackets and reheaters by gauge............. lbs. per sq. in.
24. Temperature of main supply of feed water.............. deg. Fahr.
25. Temperature of auxiliary supplies of feed water:
    - (*a*) .... "
    - (*b*) .... "
    - (*c*) .... "
26. Ideal feed-water temperature corresponding to pressure of steam in the exhaust pipe, allowance being made for heat derived from jacket or reheater drips. ...... "

Data Relating to Heat Measurement.

27. Heat units per pound of feed water, main supply. ..... B.t.u.
28. Heat units per pound of feed water, auxiliary supplies:
    - (*a*) .... "
    - (*b*) .... "
    - (*c*) .... "
29. Heat units consumed per hour, main supply. .... "
30. Heat units consumed per hour, auxiliary supplies:
    - (*a*) .... "
    - (*b*) .... "
    - (*c*) .... "
31. Total heat units consumed per hour for all purposes. .... "
32. Loss of heat per hour due to leakage of plant, drips, etc. . "
33. Net heat units consumed per hour:
    - (*a*) By engine alone.................................. "
    - (*b*) By auxiliaries................................... "
34. Heat units consumed per hour by engine alone, reckoned from temperature given in line 26.................. "

Indicator Diagrams.

1st Cyl. 2d Cyl. 3d Cyl.

35. Commercial cut-off in per cent of stroke........
36. Initial pressure in pounds per square inch above atmosphere ....
37. Back pressure at mid-stroke, above or below atmosphere in pounds per square inch .....
38. Mean effective pressure in lbs. per sq. in.......
39. Equivalent mean effective pressure in lbs. per sq. in.:.................
    (*a*) Referred to first cylinder........ ....
    (*b*) Referred ro second cylinder ....
    (*c*) Referred to third cylinder..... ....
40. Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves.
    Pressure above zero in lbs. per sq. in.:
    (*a*) Near cut-off ....
    (*b*) Near release.... ... .
    (*c*) Near beginning of compression. .. ..
    Percentage of stroke at points where pressures are measured:
    (*a*) Near cut-off..... ....
    (*b*) Near release .. ..
    (*c*) Near beginning of compression ...
41. Steam accounted for by indicator in pounds per I.H.P. per hour:
    (*a*) Near cut-off ....
    (*b*) Near release. . ...
42. Ratio of expansion.. ....

Speed.

43. Revolutions per minute.......... .... rev.

Power.

44. Indicated horse power developed by main-engine cylinders:
    First cylinder . ... H.P.
    Second cylinder . . .. "
    Third cylinder .... "
    Total. .. .. "
45. Brake horse power developed by engine. .. .. "

*Standard Efficiency and other Results.**

46. Heat units consumed by engine auxiliaries per hour:
    (*a*) Per indicated horse power .... B.t.u.
    (*b*) per brake horse power. .. .. "
47. Equivalent standard coal in lbs. per hour:
    (*a*) Per indicated horse power...................... lbs.
    (*b*) Per brake horse power. .. .. "
48. Heat units consumed by main engine per hour corresponding to ideal maximum temperature of feed water given in line 26, British standard:
    (*a*) Per indicated horse power.. .... B.t.u.
    (*b*) Per brake horse power . . . . . . "
49. Dry steam consumed per indicated horse power per hour:
    (*a*) Main cylinders including jackets. .... lbs.
    (*b*) Auxiliary cylinders . . .. "
    (*c*) Engine and auxiliaries . ... "
50. Dry steam consumed per brake horse power per hour:
    (*a*) Main cylinders including jackets. ... . "
    (*b*) Auxiliary cylinders . . . . "
    (*c*) Engines and auxiliaries .. ... "
51. Percentage of steam used by main-engine cylinders accounted for by indicator diagrams, near cut-off of high-pressure cylinder ... . per cent.

Additional Data.

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is used. Also give copies of indicator diagrams nearest the mean, and the corresponding scales.

TABLE NO III.

Data and Results of Feed water Test of Steam Engine.

Arranged according to the Short Form advised by the Engine Test Committee of the American Society of Mechanical Engineers. Code of 1902.

1. Made by. ... . of .. ..
   on engine located at..... ....
   to determine . ...
   ....

---

* The horse power referred to above (items 46–50) is that of the main engine, exclusive of auxiliaries.

2. Date of trial ....
3. Type of engine (simple, compound, or other multiple expansion; condensing or non-condensing)..
....
4. Class of engine (mill, marine, locomotive, pumping, electric, or other)....
5. Rated power of engine .. . .
6. Name of builders .. ..
7. Number and arrangement of cylinders of engine; how lagged; type of valves and of condensers..... . . ..

	1st Cyl.	2d Cyl.	3d Cyl.
8. Dimensions of engine			
(*a*) Single or double acting.................			
(*b*) Cylinder dimensions:			
Bore........ . . ..in.			
Stroke. . .. . ft.			
Diameter of piston rod............ in.			
Diameter of tail rod .. . . in.			
(*c*) Clearance in per cent of volume displaced by piston per stroke:			
Head end			
Crank end...			
Average.			
(*d*) Ratio of volume of each cylinder to volume of high-pressure cylinder			
(*e*) Horse-power constant for one pound mean effective pressure, and one revolution per minute......			

Total Quantities, Time, Etc.

9. Duration of test..	hours.
10. Water fed to boilers from main source of supply........	lbs.
11. Water fed from auxiliary supplies:	
(*a*)	"
(*b*)	"
(*c*)	"
12. Total water fed from all sources..	"
13. Moisture in steam or superheating near throttle *........	p. c. or deg.
14. Factor of correction for quality of steam.	
15. Total dry steam consumed for all purposes.... . . .	lbs.

---

* In case of superheated steam engines, determine, if practicable, the temperature of the steam in each cylinder.

Hourly Quantities.

16. Water fed from main source of supply.............. ... **lbs.**
17. Water fed from auxiliary supplies:
    (*a*) .... “
    (*b*) .... “
    (*c*) .... “
18. Total water fed to boilers per hour .... “
19. Total dry steam consumed per hour.............. .... . “
20. Loss of steam and water per hour due to leakage of plant, drips, etc...... . . .. “
21. Net dry steam consumed per hour by engine and auxiliaries “
22. Dry steam consumed per hour:
    (*a*) Main cylinders..... .... “
    (*b*) Jackets and reheaters . ... “

Pressures and Temperatures (Corrected).

23. Steam-pipe pressure near throttle, by gauge.. .... lbs. per sq. in.
24. Barometric pressure of atmosphere in inches of mercury . ins.
25. Pressure in first receiver by gauge . ..... lbs. per sq. in.
26. Pressure in second receiver by gauge. . . . . “
27. Vacuum in condenser:
    (*a*) In inches of mercury . . .. ins.
    (*b*) Corresponding total pressure . ... lbs. per sq. in.
28. Pressure in steam jackets by gauge ... .. . “
29. Pressure in reheater by gauge . .... “
30. Superheating of steam in first receiver . . . .. deg. Fahr
31. Superheating of steam in second receiver... . ... “

Indicator Diagrams.

1st Cyl. 2d Cyl. 3d Cyl.

32. Commercial cut-off in per cent of stroke .
33. Initial pressure in lbs. per sq. in. above atmosphere...... . . ..
34. Back-pressure at mid-stroke above or below atmosphere in lbs. per sq. in..... . ..
35. Mean effective pressure in lbs. per sq. in.. . . . .
36. Equivalent mean effective pressure in lbs. per sq. in. per indicated H.P:
    (*a*) Referred to first cylinder . . ..
    (*b*) Referred to second cylinder . .. ...
    (*c*) Referred to third cylinder. ....

1st Cyl. 2d Cyl. 3d Cyl.

37. Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves..........

Pressures above zero in lbs. per sq. in.:

(*a*) Near cut-off..........

(*b*) Near release..........

(*c*) Near beginning of compression ....

Percentage of stroke at points where pressures are measured:

(*a*) Near cut-off..........

(*b*) Near release ....

(*c*) Near beginning of compression ....

38. Aggregate M.E.P. in lbs. per sq. in. referred to each cylinder given in heading ......

39. Mean back-pressure above zero, lbs. per sq. in...

40. Steam accounted for in lbs. per indicated horse power per hour:

(*a*) Near cut-off ......

(*b*) Near release ....

41. Ratio of expansion:

(*a*) Commercial ....

(*b*) Ideal..........

Speed.

42. Revolutions per minute .... rev.

43. Piston speed per minute .... ft.

Power.

44. Indicated horse power developed by main engine cylinders:

First cylinder .... H.P.

Second cylinder .... "

Third cylinder .... "

Total.......... "

Efficiency Results.

45. Dry steam consumed per indicated H.P. per hour:

(*a*) Main cylinder, including jackets .... lbs.

(*b*) Auxiliary cylinders, etc. .... "

(*c*) Engine and auxiliaries.......... "

46. Percentage of steam used by main engine cylinders accounted for by indicator diagrams:

	1st Cyl.	2d Cyl.	3d Cyl.
(*a*) Near cut-off........................			
(*b*) Near release........................			

Sample Diagrams.

Copies of indicator diagrams, nearest the mean, with corresponding scales, should be given in connection with table.

DEPARTMENT OF EXPERIMENTAL ENGINEERING, SIBLEY COLLEGE.

JACKET STEAM, DATA AND RESULT

Test of Engine built by..

Tested at.. *Date*................

	Cu Ft. Per Hr.	Temp.	Wt Per Cu Ft.	Weight Per Hr
High-pressure Cylinder				
Intermediate "				
Low-pressure "				
First Receiver				
Second Receiver				
Total				

Measurements made by

....

Engine tested by

....

....

....

....

	H. P. Cylinder			First Receiver			Int. Cylinder.			Second Receiver.			L P Cylinder.		
	Time	Temp	Cu Ft	Time	Temp	Cu. Ft.	Time.	Temp	Cu Ft	Time	Temp.	Cu. Ft	Time.	Temp.	Cu Ft.

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Test of..............................*Engine. Built by*.. *Tested at*................

HIGH-PRESSURE CYLINDER	INTERMEDIATE CYLINDER	LOW-PRESSURE CYLINDER	...................191 By.................
Clearance, head end, per cent.......	Clearance, head end, per cent.....	Clearance, head end per cent.....	....
" crank " "	" crank " "	" crank " "	....
Scale of spring................	Scale of spring..................	Scale of spring..................	....
Dia. of piston, inches............	Dia. of piston, inches..........	Dia. of piston, inches..........	....
" " rod, inches.......	" " rod, inches.........	" " rod, inches.........	....
Length of stroke, feet..........	Length of stroke, feet	Length of stroke, feet.........	....
" brake arm, feet	" brake arm, feet.....	" brake arm, feet.........	....
Brake load; gross..., tare...net	Brake load; gross , tare. .net	Brake load; gross, .. tare...net..	Barometer.......In. Hg.

		REVOLUTIONS		PRESSURES					TEMPERATURES								WEIGHTS						VOL.
				Lbs. by Gauge			Inches Mercury										Condensed Steam			Separating Cal.*			
	Time.	R.P.M.	Contin. Counter.	Steam.	1st Receiver.	2d Receiver.	Condenser.	Back Press. in Throttling Cal	Steam.	1st Receiver.	2d Receiver.	Condensed Steam.	Injection Water	Discharge Water.	Throttling Calorimeter.	Engine Room.	Tare.	Gross.	Net.	Tare.	Gross.	Net.	Cu. Ft. of Condensing Water.
1																							
2																							
3																							
4																							
Average																							

* In exhaust of H.P. cylinder.

SIBLEY COLLEGE, CORNELL UNIVERSITY.

Cards from Engine built by..

Tested at..*Date*.................................

Name of Planimeter used..................................*No*.........................

Cards integrated by..................................*Results computed by*.................

Checked by..

Scale of Spring..................................*Av. Diagram Water Rate Cut-off*..............

Release..................

HEAD END.											CRANK END.									
					Absolute Pressure.										Absolute Pressure.					
No	Area.	Length.	M.E.P.	I H.P	End of Compression.	Cut-off.	Release.	Back	Cut-off % Stroke.	Release % Stroke.	Area.	Length.	M E P	I.H.P.	End of Compression.	Cut-off.	Release.	Back.	Cut-off % Stroke.	Release % Stroke.

363. The Testing of Pumping Engines (Steam Pumps). — Considering a pumping engine merely as a steam engine, in which the useful work is represented by the water pumped against certain resistances, it will be seen that there should be no great difference between the methods of testing employed for the two types of engine, at least as far as the steam end of the plant is concerned. To determine the useful performance of the pump, certain measurements in connection with the pump cylinders become necessary. These should include the taking of indicator cards, to determine the water horse power, the measurement of the water pumped, and of the suction and discharge heads, and the determination of slip or leakage of plungers and valves.

The method of taking indicator cards from the pump cylinders is the same as for steam cylinders (see Chapter XV), and the water horse power is determined by the same method of computation. The ratio between the water horse power and the I.H.P. of the steam cylinders may be considered as representing the mechanical efficiency of the pump.

The total head h pumped against is equal to the sum of the following heads:

$$h = \left[h_d - h_s + h_1 + \frac{v_d^2 - v_s^2}{2g}\right] \text{ feet,}$$

in which

h_d = discharge head. This is usually measured by connecting a gauge to the discharge main close to the pump.

h_s = suction head. Usually measured by connecting a gauge or mercury manometer to the suction main close to the pump. The usual case is to have the pump lift the liquid. In that case h_s must be considered negative and this factor in the equation then becomes $-(-h_s) = +h_s$.

h_1 = the vertical distance in feet between the discharge and suction gauges, or between the center of the discharge gauge and the point of attachment of the mercury manometer.

v_d and v_s are the velocities in feet per second in the discharge and suction mains respectively at the points of gauge connection. If the suction and discharge mains are of the same size, this factor of course cancels out.

The total head pumped against in connection with the weight of water pumped, determines the foot-pounds of useful work done by the pump.

The determination of the actual amount of water pumped (pump delivery *) can rarely be done by direct weighing, on account of the quantity usually involved, but must in nearly all cases be done by some indirect means. For this purpose weirs, nozzles, and Venturi meters have all been proposed and used. The Committee, however, recommends the method of using the plunger displacement corrected for leakage (slip) of plungers and valves, on the ground that the use of the other schemes mentioned involves the assumption of coefficients which may make them no more accurate than the method recommended. The latter certainly has the advantage of great simplicity. For directions concerning the determination of leakage, see the provisions of the Code.

The term *duty* is often used as an efficiency standard in connection with pumping machinery. It refers to the number of foot-pounds of work done by a pump for a certain energy quantity supplied to the machine. Thus we speak of duty per 100 pounds of coal, or duty per 1000 pounds of steam, or duty per 1,000,000 B.t.u. The Committee recommends the use of the term duty in connection with the heat-unit basis stated in preference to the first two definitions given, for the reason that 100 pounds of coal may mean very different quantities of heat supply for different parts of the country, and 1000 pounds of steam will represent different heat quantities, depending upon the conditions of the steam. The 1,000,000 B.t.u. basis, on the other hand, is a fixed basis. How the total heat supply to the engine is to be computed is shown in the following formulas, which show the method of computation for the principal results for a steam-pump test. These are quoted directly from the Code.

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* A distinction should be made here between capacity and delivery. The former is always based on plunger displacement.

1. $$\text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat units consumed}} \times 1,000,000$$
$$= \frac{A\,(P \pm p + s) \times L \times N}{H} \times 1,000,000 \text{ (foot-pounds).}$$

2. $$\text{Percentage of leakage} = \frac{C \times 144}{A \times L \times N} \times 100 \text{ (per cent).}$$

3. Capacity = number of gallons of water discharged in 24 hours
$$= \frac{A \times L \times N \times 7.4805 \times 24}{D \times 144}$$
$$= \frac{A \times L \times N \times 1.24675}{D} \text{ (gallons).}$$

4. Percentage of total frictions
$$= \left[\frac{I.H.P. - \dfrac{A\,(P \pm p + s) \times L \times N}{D \times 60 \times 33,000}}{I.H.P.}\right] \times 100$$
$$= \left[1 - \frac{A\,(P \pm p + s) \times L \times N}{A_s \times M.E.P. \times L_s \times N_s}\right] \times 100 \text{ (per cent);}$$

or, in the usual case, where the length of the stroke and number of strokes of the plunger are the same as that of the steam-piston, this last formula becomes:

$$\text{Percentage of total frictions} = \left[1 - \frac{A\,(P \pm p + s)}{A_s \times M.E.P.}\right] \times 100 \text{ (per cent).}$$

In these formulas the letters refer to the following quantities:

A = Area, in square inches, of pump plunger or piston, corrected for area of piston-rod. (When one rod is used at one end only, the correction is one-half the area of the rod. If there is more than one rod, the correction is multiplied accordingly.)

P = Pressure, in pounds per square inch, indicated by the gauge on the force main.

p = Pressure, in pounds per square inch, corresponding to indication of the vacuum gauge on suction main (or pressure gauge, if the suction pipe is under a head). The indication of the vacuum gauge, in inches of mercury, may be converted into pounds by dividing it by 2.035.

s = Pressure, in pounds per square inch, corresponding to distance between the centers of the two gauges. The computation for this pressure is made by multiplying the distance, expressed in feet, by the weight of one cubic foot of water at the temperature of the pump well, and dividing the product by 144.

L = Average length of stroke of pump plunger, in feet.

N = Total number of single strokes of pump plunger made during the trial.

A_s = Area of steam cylinder, in square inches, corrected for area of piston-rod. The quantity $A_s \times M.E.P.$ in an engine having more than one cylinder is the sum of the various quantities relating to the respective cylinders.

L_s = Average length of stroke of steam piston, in feet.

N_s = Total number of single strokes of steam piston during trial.

$M.E.P.$ = Average mean effective pressure, in pounds per square inch, measured from the indicator-diagrams taken from the steam cylinder.

$I.H.P.$ = Indicated horse power developed by the steam cylinder.

C = Total number of cubic feet of water which leaked by the pump plunger during the trial, estimated from the results of the leakage test.

D = Duration of trial, in hours.

H = Total number of heat units [B.t.u.] consumed by engine = weight of water supplied to boiler by main feed-pump × total heat of steam of boiler pressure reckoned from temperature of main feed-water + weight of water supplied by jacket-pump × total heat of steam of boiler pressure reckoned from temperature of jacket-water + weight of any other water supplied × total heat of steam reckoned from its temperature of supply. The total heat of the steam is corrected for the moisture or superheat which the steam may contain. For moisture the correction is subtracted, and is found by multiplying the latent heat of the steam by the percentage of moisture, and dividing the product by 100. For superheat, the correction is added, and is found by multiplying the number of degrees of superheating (i.e., the excess of the temperature of the steam above the normal temperature of saturated steam) by the specific heat. No allowance is made for heat added to the feed-water, which is derived from any source, except the engine or some accessory of the engine Heat added to the water by the use of a flue heater at the boiler is not to be deducted Should heat be abstracted from the flue by means of a steam reheater connected with the intermediate receiver of the engine, this heat must be included in the total quantity supplied by the boiler.

STANDARD METHOD OF CONDUCTING DUTY TRIALS OF PUMPING ENGINES.

1. Test of Feed-water Temperatures.

The plant is subjected to a preliminary run, under the conditions determined upon for the test, for a period of three hours, or such a time as is necessary to find the temperature of the feed-water (or the several temperatures, if there is more than one supply) for use in the calculation of the duty. During this test observations of the temperature are made every fifteen minutes. Frequent observations are also made of the speed, length of stroke, indication of water-pressure gauges, and other instruments, so as to have a record of the general conditions under which this test is made

DIRECTIONS FOR OBTAINING FEED-WATER TEMPERATURES.

When the feed-water is all supplied by one feeding instrument, the temperature to be found is that of the water in the feed-pipe near the point where it enters the boiler. If the water is fed by an injector this temperature is to be corrected for the heat added to the water by the injector, and for this purpose the temperatures of the water entering and of that leaving the injector are both observed. If the water does not pass through a heater on its way to the boiler (that is, that form of heater which depends upon the rejected heat of the engine, such as that contained in the exhaust steam either of the main cylinders or of the auxiliary pumps) it is sufficient, for practical purposes, to take the temperature of the water at the source of supply, whether the feeding instrument is a pump or an injector.

When there are two independent sources of feed-water supply, one the main supply from the hot well, or from some other source, and the other an auxiliary supply derived from the water condensed in the jackets of the main engine and in the live-steam reheater, if one be used, they are to be treated independently. The remarks already made apply to the first, or main, supply. The temperature of the auxiliary supply, if carried by an independent pipe either direct to the boiler or to the main feed-pipe near the boiler, is to be taken at a convenient point in the independent pipe.

When a separator is used in the main steam-pipe, arranged so as to discharge the entrained water back into the boiler by gravity, no account need be made of the temperature of the water thus returned. Should it discharge either into the atmosphere to waste, to the hot well, or to the jacket tank, its temperature is to be determined at the point where the water leaves the separator before its pressure is reduced.

When a separator is used, and it drains by gravity into the jacket tank, this tank being subjected to boiler pressure, the temperatures of the separator water and jacket water are each to be taken before their entrance to the tank.

Should there be any other independent supply of water, the temperature of that is also to be taken on this preliminary test.

DIRECTIONS FOR MEASUREMENT OF FEED-WATER.

As soon as the feed-water temperatures have been obtained the engine is stopped, and the necessary apparatus arranged for determining the weight of the feed-water consumed, or of the various supplies of feed-water, if there is more than one.

In order that the main supply of feed-water may be measured, it will generally be found desirable to draw it from the cold-water service main. The best form of apparatus for weighing the water consists of two tanks, one of which rests upon a platform scale supported by staging, while the other is placed underneath. The water is drawn from the service main into the upper tank, where it is weighed, and it is then emptied into the lower tank. The lower tank serves as a reservoir, and to this the suction-pipe of the feeding apparatus is connected.

The jacket water may be measured by using a pair of small barrels, one being filled while the other is being weighed and emptied. This water, after being measured, may be thrown away, the loss being made up by the main feed pump. To prevent evaporation from the water, and consequent loss on account of its highly heated condition, each barrel should be partially filled with cold water previous to using it for collecting the jacket water, and the weight of this water treated as tare.

When the jacket water drains back by gravity to the boiler, waste of live steam during the weighing should be prevented by providing a small vertical chamber, and conducting the water into this receptacle before its escape. A glass water-gauge is attached so as to show the height of water inside the chamber, and this serves as a guide in regulating the discharge valve. The chamber may be made of piping in the manner shown in the appended figure (501).

When the jacket water is returned to the boiler by means of a pump, the discharge valve should be throttled during the test, so that the pump may work against its usual pressure; that is, the boiler pressure as nearly as may be, a gauge being attached to the discharge pipe for this purpose.

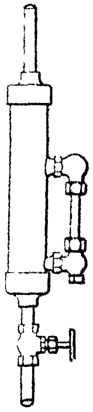
FIG. 501.

When a separator is used and the entrained water discharges either to waste, to the hot well, or to the jacket tank, the weight of this water is to be determined, the water being drawn into barrels in the manner pointed out for measuring the jacket water. Except in the case where the separator discharges into the jacket tank, the entrained water thus found is treated, in the calculations, in the same manner as moisture shown by the calorimeter test. When it discharges into the jacket tank, its weight is simply subtracted from the total weight of water fed, and allowance made for heat of this water lost by radiation between separator and tank.

When the jackets are drained by a trap, and the condensed water goes either to waste or to the hot well, the determination of the quantity used is not necessary to the main object of the duty trial, because the main feed-pump in such cases supplies all the feed-water. For the sake of having complete data, however it is desirable that this water be measured, whatever the use to which it is applied.

Should live steam be used for reheating the steam in the intermediate receiver, it is desirable to separate this from the jacket steam, if it drains into the same tank, and measure it independently. This, likewise, is not essential to the main object of the duty trial, though useful for purposes of information.

The remarks as to the manner of preventing losses of live steam and of evaporation, in the measurement of jacket water, apply to the measurement of any other hot water under pressure which may be used for feed-water.

Should there be any other independent supply of water to the boiler besides those named, its quantity is to be determined independently, apparatus for all these measurements being set up during the interval between the preliminary run and the main trial, when the plant is idle.

2. THE MAIN DUTY TRIAL.

The duty trial is here assumed to apply to a complete plant, embracing a test of the performance of the boiler as well as that of the engine The test of the two will go on simultaneously after both are started, but the boiler test will begin a short time in advance of the beginning of the engine test, and continue a short time after the engine test is finished The mode of procedure is as follows:

The plant having been worked for a suitable time under normal conditions, the fire is burned down to a low point and the engine brought to rest. The fire remaining on the grate is then quickly hauled, the furnace cleaned, and the refuse withdrawn from the ash pit The boiler test is now started, and this test is made in accordance with the rules for a standard method recommended by the Committee on Boiler Tests of the American Society of Mechanical Engineers * This method, briefly described, consists in starting the test with a new fire lighted with wood, the boiler having previously been heated to its normal working degree; operating the boiler in accordance with the conditions determined upon, weighing coal, ashes, and feed-water, observing the draught, temperatures of feed-water and escaping gases, and such other data as may be incidentally desired; determining the quantity of moisture in the coal and in the steam; and at the close of the test hauling the fire, and deducting from the weight of coal fired whatever unburned coal is contained in the refuse withdrawn from the furnace, the quantity of water in the boiler and the steam pressure being the same as at the time of lighting the fire at the beginning of the test.

Previous to the close of the test it is desirable that the fire should be burned down to a low point, so that the unburned coal withdrawn may be in a nearly consumed state. The temperature of the feed-water is observed at the point where the water leaves the engine heater, if this be used, or at the point where it enters the flue heater, if that apparatus be employed. Where an injector

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* Vol. VI, p. 267, Transactions A. S. M. E., 1885. (See also Chap. XVII.)

is used for supplying the water, a deduction is to be made in either case for the increased temperature of the water derived from the steam which it consumes.

As soon after the beginning of the boiler test as practicable the engine is started and preparations are made for the beginning of the engine test. The formal beginning of this test is delayed till the plant is again in normal working condition, which should not be over one hour after the time of lighting the fire. When the time for beginning arrives the feed-water is momentarily shut off, and the water in the lower tank is brought to a mark. Observations are then made of the number of tanks of water thus far supplied, the height of water in the gauge-glass of the boiler, the indication of the counter on the engine, and the time of day, after which the supply of feed-water is renewed, and the regular observations of the test, including the measurement of the auxiliary supplies of feed-water, are begun. The engine test is to continue at least ten hours. At its expiration the feed-pump is again momentarily stopped, care having been taken to have the water slightly higher than at the start, and the water in the lower tank is brought to the mark. When the water in the gauge-glass has settled to the point which it occupied at the beginning, the time of day and the indication of the counter are observed, together with the number of tanks of water thus far supplied, and the engine test is held to be finished. The engine continues to run after this time till the fire reaches a condition for hauling, and completing the boiler test. It is then stopped, and the final observations relating to the boiler test are taken.

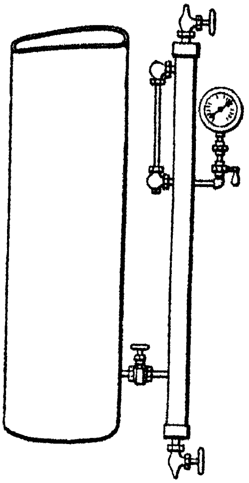
FIG. 502. — METHOD OF OBTAINING STEADY GAUGE READINGS.

The observations to be made and data obtained for the purposes of the engine test, or duty trial proper, embrace the weight of feed-water supplied by the main feeding apparatus, that of the water drained from the jackets, and any other water which is ordinarily supplied to the boiler, determined in the manner pointed out. They also embrace the number of hours duration, and number of single strokes of the pump during the test; and, in direct-acting engines, the length of the stroke; together with the indications of the gauges attached to the force and suction mains, and indicator-diagrams from the steam cylinders. It is desirable that pump diagrams also be obtained.

Observations of the length of stroke, in the case of direct-acting engines, should be made every five minutes; observations of the water-pressure gauges every fifteen minutes; observations of the remaining instruments, such as steam gauge, vacuum gauge, thermometer in pump well, thermometer in feed-pipe, thermometer showing temperature of engine room, boiler room, and outside air, thermometer in flue, thermometer in steam pipe, if the boiler has steam-heating surface, barometer, and other instruments which may be used, every half-hour. Indicator-diagrams should be taken every half-hour.

When the duty trial embraces simply a test of the engine, apart from the boiler, the course of procedure will be the same as that described, excepting that the fires will not be hauled, and the special observations relating to the performance of the boiler will not be taken.

DIRECTIONS REGARDING ARRANGEMENT AND USE OF INSTRUMENTS, AND OTHER PROVISIONS FOR THE TEST.

The gauge attached to the force main is liable to a considerable amount of fluctuation unless the gauge-cock is nearly closed. The practice of choking the cock is objectionable. The difficulty may be satisfactorily overcome, and a nearly steady indication secured, with cock wide open, if a small reservoir having an air-chamber is interposed between the gauge and the force main, in the manner shown in the appended figure (502). By means of a gauge-glass on the side of the chamber and an air-valve, the average water-level may be adjusted to the height of the center of the gauge, and correction for this element of variation is avoided. If not thus adjusted, the reading is to be referred to the level shown, whatever this may be.

To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water cylinders, in a position parallel to the piston-rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two ends of the stroke, are thus readily observed, and the distance moved over computed. If the length of the stroke can be determined by the use of some form of registering apparatus, such a method of measurement is preferred. The personal errors in observing the exact marks, which are liable to creep in, may thereby be avoided.

The form of calorimeter to be used for testing the quality of the steam is left to the decision of the person who conducts the trial. It is preferred that some form of continuous calorimeter be used, which acts directly on the moisture tested. If either the superheating calorimeter * or the wire-drawing † instrument be employed, the steam which it discharges is to be measured either by numerous short trials, made by condensing it in a barrel of water previously weighed, thereby obtaining the rate by which it is discharged, or by passing it through a surface condenser of some simple construction, and

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* Vol. VII, p. 178, 1886, Trans A. S. M. E.

† Vol. XI, p. 791, 1890, Transactions A. S. M E (Paper on "A Universal Calorimeter," May, 1890.) (See Chap. XIV.)

measuring the whole quantity consumed. When neither of these instruments is at hand, and dependence must be placed upon the barrel calorimeter, scales should be used which are sensitive to a change in weight of a small fraction of a pound, and thermometers which may be read to tenths of a degree. The pipe which supplies the calorimeter should be thoroughly warmed and drained just previous to each test. In making the calculations the specific heat of the material of the barrel or tank should be taken into account, whether this be of metal or of wood.

If the steam is superheated, or if the boiler is provided with steam-heating surface, the temperature of the steam is to be taken by means of a high-grade thermometer resting in a cup holding oil or mercury, which is screwed into the steam pipe so as to be surrounded by the current of steam. The temperature of the feed-water is preferably taken by means of a cup screwed into the feed-pipe in the same manner.

Indicator pipes and connections used for the water cylinders should be of ample size, and, as far as possible, free from bends. Three-quarter-inch pipes are preferred, and the indicators should be attached one at each end of the cylinder. It should be remembered that indicator springs which are correct under steam heat are erroneous when used for cold water. When such springs are used, the actual scale should be determined, if calculations are made of the indicated work done in the water cylinders. The scale of steam springs should be determined by a comparison, under steam pressure, with an accurate steam gauge at the time of the trial, and that of water springs by cold dead-weight test.

The accuracy of all the gauges should be carefully verified by comparison with a reliable mercury column.* Similar verification should be made of the thermometers, and if no standard is at hand, they should be tested in boiling water and melting ice.

To avoid errors in conducting the test, due to leakage of stop-valves either on the steam pipes, feed-water pipes, or blow-off pipes, all these pipes not concerned in the operation of the plant under test should be disconnected.

3. LEAKAGE TEST OF PUMP.

As soon as practicable after the completion of the main trial (or at some time immediately preceding the trial), the engine is brought to rest, and the rate determined at which leakage takes place through the plunger and valves of the pump, when these are subjected to the full pressure of the force main.

The leakage of an inside plunger [the only type which requires testing] is most satisfactorily determined by making the test with the cylinder head removed. A wide board or plank may be temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water in the manner of a dam, and an opening made in the temporary head thus provided for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or, if this position is not practicable, at the end of the stroke), and the water from the force main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured.

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* AUTHOR'S NOTE. See the chapter on the Measurement of Pressure for other apparatus fully as accurate and easier to operate.

Should the escape of the water into the engine room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or orifice taking the place of the overflow pipe in the wooden head. The apparatus may be constructed, if desired, in a somewhat rude manner, and yet be sufficiently accurate for practical requirements. The test should be made, if possible, with the plunger in various positions.

In the case of a pump so planned that it is difficult to remove the cylinder head, it may be desirable to take the leakage from one of the openings which are provided for the inspection of the suction valves, the head being allowed to remain in place.

It is here assumed that there is a practical absence of valve leakage, a condition of things which ought to be attained in all well-constructed pumps. Examination for such leakage should be made first of all, and if it occurs and is found to be due to disordered valves, it should be remedied before making the plunger test. Leakage of the discharge valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction valves will be shown by the disappearance of water which covers them.

If valve leakage is found which cannot be remedied, the quantity of water thus lost should also be tested. The determination of the quantity which leaks through the suction valves, where there is no gate in the suction pipe, must be made by indirect means. One method is to measure the amount of water required to maintain a certain pressure in the pump cylinder when this is introduced through a pipe temporarily erected, no water being allowed to enter through the discharge valves of the pump.

The exact methods to be followed in any particular case, in determining leakage, must be left to the judgment and ingenuity of the person conducting the test.

4. TABLE OF DATA AND RESULTS.

In order that uniformity may be secured, it is suggested that the data and results, worked out in accordance with the standard method, be tabulated in the manner indicated in the following scheme:

DUTY TRIAL OF ENGINE.

Dimensions.

1. Number of steam cylinders..........................
2. Diameter of steam cylinders.......................... ins.
3. Diameter of piston-rods of steam cylinders.............. "
4. Nominal stroke of steam-pistons...................... ft.
5. Number of water-plungers..........................
6. Diameter of plungers .. .. ins.
7. Diameter of piston-rods of water cylinders.... .... "
8. Nominal stroke of plungers .. .. ft.
9. Net area of plungers .... ins.
10. Net area of steam pistons............ .. .. "
11. Average length of stroke of steam pistons during trial.... ft.
12. Average length of stroke of plungers during trial...... ... "

(Give also complete description of plant.)

Temperatures.

13. Temperature of water in pump well.... .... degs.
14. Temperature of water supplied to boiler by main feed-pump ... . "
15. Temperature of water supplied to boiler from various other sources.......... .... "

Feed-water.

16. Weight of water supplied to boiler by main feed-pump... lbs.
17. Weight of water supplied to boiler from various other sources.... .... "
18. Total weight of feed-water supplied from all sources...... "

Pressures.

19. Boiler pressure indicated by gauge.................. ..... "
20. Pressure indicated by gauge on force main............. "
21. Vacuum indicated by gauge on suction main........ ... ins.
22. Pressure corresponding to vacuum given in preceding line. lbs.
23. Vertical distance between the centers of the two gauges.. ins.
24. Pressure equivalent to distance between the two gauges.. lbs.

Miscellaneous Data.

25. Duration of trial .... hrs.
26. Total number of single strokes during trial... ....
27. Percentage of moisture in steam supplied to engine, or number of degrees of superheating. .... p. c. or deg.
28. Total leakage of pump during trial, determined from results of leakage test .... lbs.
29. Mean effective pressure, measured from diagrams taken from steam cylinders............................. M.E.P

Principal Results.

30. Duty	ft.-lbs.
31. Percentage of leakage	per cent.
32. Capacity	gals.
33. Percentage of total frictions	per cent.

*Additional Results.**

34. Number of double strokes of steam piston per minute	
35. Indicated horse power developed by the various steam cylinders	I.H.P.
36. Feed-water consumed by the plant per hour	lbs.
37. Feed-water consumed by the plant per indicated horse power per hour, corrected for moisture in steam	"
38. Number of heat units consumed per indicated horse power per hour	B.t.u.
39. Number of heat units consumed per indicated horse power per minute	"
40. Steam accounted for by indicator at cut-off and release in the various steam cylinders	lbs.
41. Proportion which steam accounted for by indicator bears to the feed water consumption	

Sample Diagrams Taken From Steam Cylinders.

[Also, if possible, full measurements of the diagrams, embracing pressures at the initial point, cut-off, release, and compression; also back pressure, and the proportions of the stroke completed at the various points noted.]

42. Number of double strokes of pump per minute	
43. Mean effective pressure, measured from pump diagrams	M.E.P.
44. Indicated horse power exerted in pump cylinders	I.H.P.
45. Work done (or duty) per 100 lbs. of coal	ft.-lbs.

Sample Diagrams Taken From Pump Cylinders.

DATA AND RESULTS OF BOILER TEST.

For the items of this table see the Boiler Code, Chap. XVII, of this book.

364. The Testing of Locomotives.—Tests on locomotives are divided into *shop tests* and *road tests*. In the former the engine is supported, so that, although it may be operated at full speed as far as revolution of drivers is concerned, the locomotive as a whole has no motion. The equipment required for such a test is rather elaborate and quite costly, and hence out of reach of the ordinary testing laboratory. Several of such testing stands are, however, found in one or two of the colleges and in the shops of some of the important railroads. The road tests, as the name implies, are made with the locomotive in actual service drawing a train over the road. The advantage of shop tests over road tests consists in the fact that it is possible to make measurements with probably somewhat greater accuracy and that all conditions can be maintained constant. This is important when it is desired to study one particular feature of operation. The road tests, on the other hand, exhibit actual operating results and therefore furnish a valuable complement to the shop test. When engine performance alone, as compared with that of other engines, is the matter under investigation, it is desirable to make the road test of considerable length, say not under 100 miles, and to have each engine draw the same train, which should be a special rather than a regular train, in order to be independent as far as number of stops is concerned.

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* These are not necessary to the main object, but it is desirable to give them.

The Committee recommends that a dynamometer car be used in all road tests in order to determine accurately the useful work done. The Code gives several figures illustrating the arrangement of apparatus in a dynamometer car, but, since no description is given, these figures are of doubtful value for the purpose of this book. The student is therefore referred to articles dealing with the operation of dynamometer cars as well as with the construction of the testing stands for shop tests which have appeared from time to time in various technical railway journals.

The Committee also recommends the adoption of an *efficiency standard* for locomotives, defining it as the number of pounds of "standard coal" consumed per dynamometer horse power per hour. Standard coal is defined as one which will, when tested in the oxygen calorimeter, show a heating value of 12,500 B.t.u per pound. To convert the actual amount of coal used into the equivalent weight of "standard coal," the weight of coal burned per dynamometer horse power per hour is multiplied by its heating value and divided by 12,500.

STANDARD METHOD OF CONDUCTING EFFICIENCY TESTS OF LOCOMOTIVES.

1. Preparations for Test, and Location of Instruments.*

A. The locomotive should be put in good condition preparatory to the test. The boiler and tubes should be tight, and both the interior and exterior surfaces should be clean, and, if possible, free from scale. There should be no lost motion in the valve gear, and the valves should be set properly. No change in the engines should be allowed during the progress of a series of tests, unless so ordered for the purposes of the trial.

A glass water-gauge should be fitted to the boiler, if not already provided. A rod should be attached to the reversing lever and carried forward to the front end of the boiler, where a graduated scale is provided and suitably marked, so that the position of the reversing lever can be seen at a glance by the person taking indicator-diagrams. The throttle-valve lever should be provided with a scale, to show the degree of opening of the throttle.

B. The valves and pistons should be tested for leakage with the engine at rest. The steam valve can be tried by setting the engine so that the valve on one side will be at the center of its throw, in which position both ports are usually covered, and pulling open the throttle valve, blocking the drivers if there is a tendency for the engine to be set in motion. Leakage of the valve, if any occurs, will show itself by escaping at the smokestack, or at the open drain-cocks. The tightness of the piston may be tested by setting the engine so that it takes steam, blocking th drivers and opening the throttle valve. This should be tried first on one cylinder and then on the other, and, if desired, it may be tried with the pistons at various points in the stroke. The leakage, if any occurs, will be shown either at the top of the smokestack or at the open drain-cock.

C. The following instruments should be verified, or calibrated: steam gauges, draft gauge, pyrometer, thermometers for calorimeter and feed-water, water meter, tank, revolution counter, indicator springs, dynamometer springs, and dynamometer recording mechanism. The radiation loss on the steam calorimeter should be determined, or the normal readings ascertained; † and the quantity of steam which passes through the instrument in a given time should be measured.

D. The quantities of steam consumed by the air-pump, the blower, and the whistle, under conditions of common use, should be determined, thereby obtaining data by which to correct for the steam thus used. This can best be determined for each one by observing the fall of water in the gauge glass when no steam is drawn from the boiler for any other purpose; the quantity being computed from the data thus found, and the dimensions of the boiler. The leakage of the boiler should also be found, using the same method.

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* The directions here given apply largely to both shop and road tests, but especially to the latter.

† Transactions A. S. M. E., Vol. XI, p. 793.

E. To facilitate the measurement of coal and the determination of the quantity used during any desired period of the run, it is desirable to provide a sufficient number of sacks of a size holding a weight of, say 100 lbs., and weigh the coal into these sacks preparatory to starting on the test. If desired, the sacks may be numbered, to facilitate the accuracy of record.

F. The instruments and other apparatus that should be provided, and their locations, are as follows:

To facilitate the work of operating the indicators and reading the instruments at the front end, the smoke box should be surrounded with a wooden fence, or "pilot box," as it may be called, resting on the top of the cow-catcher and extending back far enough to inclose also the sides of the cylinders. This box is floored over, and the inclosure thus provided forms a convenient place for the accommodation of the assistants at this end of the locomotive, and it affords them some measure of protection against wind and rain, as also the jolting and vibrations due to rapid travel.

A special steam gauge with a long siphon is to be used for registering the boiler pressure. It can best be located on the left-hand side of the cab.

The indicator apparatus which is most suitable consists of a three-way cock * for the attachment of the indicators, and some form of pantograph motion for the driving rig. The pipes leading from the cock to the cylinder should be three-fourths inch diameter inside, and they should connect into the side of the cylinder rather than into the two heads. The indicator should also be piped so that a steam chest diagram can be drawn by it. Sharp bends in the pipe should be avoided, and they should be well covered, to intercept radiation. The three-way cock should be provided with a clamp rigidly secured to the cylinder, and thus overcome any tendency of the indicators to move longitudinally with reference o the driving rig. Absolute rigidity is highly essential in this particular. (For forms of driving rig for the indicators, see Chap. XV of this book.)

A draft gauge consisting of a U tube containing water, properly graduated in inches, should be connected to the smoke box and attached to the side of the pilot box.

A pyrometer for showing the temperature of the escaping gases should be used in a position below the tip of the exhaust nozzles.

The calorimeter should be attached either to the steam dome at a point close to the throttle opening, or to the steam passage in the saddle casting on one side, according as it is desired to obtain the character of the steam at one point or the other. The former location is preferred by the Committee. A perforated half-inch pipe should be used for sampling and conveying the steam to the calorimeter pipe. For descriptions of various forms of calorimeter which are adapted to locomotive use see Transactions A. S. M. E., Vol. X, p. 327; Vol. XII, p. 825.

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* AUTHOR'S NOTE. The use of a three-way indicator cock is not countenanced by many engineers under any circumstances.

The water meter should be attached to the suction pipe of the injector, and located at a point where it can be conveniently read when the locomotive is running. It should be provided with a check valve to prevent hot water from flowing back through it from the injector, and a strainer to intercept foreign material.

To measure the depth of the water in the tank, a metallic float should be used, carrying a vertical tube which slides upon a graduate rod, the lower end of which rests upon the bottom of the tank. This should be placed at the center of gravity of the water space. If the desired location cannot be used, provision should be made for ascertaining the level or inclination of the tank. The best device for this purpose is a plumbline of a certain length provided at the bottom with a double horizontal scale having one set of divisions parallel to the side of the tank, and the other set at right angles to it. From the readings on these scales referred to the length of the line, the level of the tank in both directions can be ascertained. A similar device should be attached to the boiler to correct for the variation in its inclination. The plumbline may be conveniently attached for this purpose at some point near the front end.

The revolution counter should be placed near the front end of the engine in plain view from the pilot box. It is operated through a belt from the driver shaft. This recommendation applies to that form of counter which shows at a glance the exact speed in revolutions per minute.

A stroke counter should be provided for showing the number of strokes made by the air-pump.

Electric connection should be made between the dynamometer car and the pilot box, so that dynamometer records and indicator-diagrams may be taken simultaneously. Another desirable provision is a speaking tube leading from the dynamometer car to the locomotive cab, and one also to the pilot box.

G. It is needless, except for a complete record of directions for preparatory work, to call attention to the desirability of having the test, and especially the road test, made under the supervision of a competent person, who is not only familiar with the details of testing, but also with the proper method of firing and the mechanical operation of the locomotive. This is a most important factor, for it is only the clear-headed and able experimenter who is likely to obtain satisfactory work in this most difficult department of engineering tests.

In the matter of assistants, the conductor of the test is best able to judge as to the number required, the various duties of the different men, and the manner of taking records. A good test can be made with eight (8) assistants, distributed in the manner indicated in the following list, which gives their duties:

Two (2) cab assistants, who read the steam gauge, the position of the throttle-valve and reversing lever, the water meter, the height of water in the tank, the height of water in the glass water-gauge, the level of the tank, the number of times the whistle is blown, the length of time the safety-valve blows, the length of time the blower is in action, the reading of the air-pump counter, the temperature of the feed-water in the tank, the time of starting and stopping the injector, the time of opening and closing the throttle-valve, and the number of sacks of coal used. These two observers have previously checked the weights of coal placed in the sacks.

Three (3) pilot-box assistants, one of whom reads the pyrometer, the draft-gauge, the steam-chest gauge, the revolution counter, and marks on the indicator diagrams the time, position of reversing lever, steam-chest pressure, and revolutions per minute. He also takes the levels of the boiler at stopping places. The other two observers are stationed at the cylinders and manipulate the indicators, one being employed on each side.

One (1) calorimeter assistant, who reads the calorimeter thermometers, and the gauges connected with the instrument, if these are employed.

Two (2) dynamometer car assistants, who record time of each start and stop, time of passing each station and each mile-post, time of taking each indicator diagram as obtained from signals of the indicator men, and all these readings are marked so far as possible on the dynamometer paper. One of these men also assists the cab observer in reading the tank depth and its levels at stopping places These men also keep a record of the direction and force of the wind, and the temperature of the atmosphere.

An additional assistant is required if the gases are sampled and analyzed.

H. It is of paramount importance, after the complete preparatory work has been accomplished, that the locomotive be subjected to a preliminary run, of sufficient duration to make a fair trial of the testing apparatus, and to give the various assistants an opportunity to become trained in their duties.

I. SHOP TEST.

A. Preparation and Location of Instruments.

In preparing for a shop test the preparations described in Section I should be followed so far as the nature of the test requires. When run as a stationary engine the locomotive is not circumscribed by the conditions of road service, and many provisions required on the road are unnecessary. It is unnecessary to determine the quantity of steam consumed by the whistle and air-pump, for these are not brought into use on the shop test, and no occasion exists for finding the quantity lost at the safety valve, for on the continuous shop run the steam pressure can be maintained at a uniform point, and blowing off readily prevented. It is unnecessary to use sacks for the convenient measure of coal, because the coal can be readily weighed up in lots as fast as needed for the test. It is unnecessary to provide a "pilot box," and no fixed location of the instruments is required, as on the road test. The feed-water may be weighed before it is supplied to the tank, and the tank may be used in this case as a reservoir, the float showing its depth. The meter would thus be unnecessary as the principal instrument of measurement, but a meter is in all cases useful as a check upon this most important element in the data. The long indicator pipes required on the road test may be dispensed with, and one indicator applied close to each end of the cylinder — a practice much to be preferred to the use of a three-way cock and the single indicator. The dynamometer car is not required, but its equivalent should be provided, consisting of a dynamometer which registers the pull on the draw-bar, in the same manner as the device used on the road.

The number of assistants required on a shop test is less than that needed for a road test. A good test can be made with four (4) assistants, distributed as follows:

One (1) assistant for operating indicators.
One (1) assistant for measuring water.
Two (2) assistants for general observations and coal measurement.
If the gases are sampled and analyzed, one more assistant is required.

B. Conditions of Test.

The test should be continued for a run of at least two (2) hours from the time normal conditions have been established.

At the close of the test the water height in the boiler, and the height of water in the tank, should be the same as at the beginning, or proper corrections made for any differences which may exist.

The fire-box and ash-pit are then cleaned, and such unburnt coal as may be contained in the refuse is separated, weighed, and deducted from the total weight of coal fired. The balance of the refuse is weighed, as also the cinders removed from the smoke box.

During the progress of the test samples of the various charges of coal should be obtained, and at its close a final sample of these should be selected, dried, and subjected to chemical analysis and calorimeter test. The weight of the sample is taken before and after drying, to ascertain the amount of moisture contained in the fuel.

C. The Data and Results.

The data and results of the shop test can best be arranged in the manner indicated in Table No. 1. So far as these are in common with the data and results obtained on the road test, the forms used on both kinds of test are identical.

TABLE NO. I.

DATA AND RESULTS OF SHOP TEST ON......ENGINE, MADE........ ..19..

General dimensions, etc. (to be accompanied by a complete description, with drawings and full dimensions).

1. Kind of engine ....
2. Size and clearance of cylinders.................................
3. Area of heating surface...................... ....
4. Area of grate surface...
5. Diameter of exhaust nozzles.....................................

		Whole Run.
	TOTAL QUANTITIES.	
6.	Duration...hrs.	
7.	Weight of dry coal burned, including .4 weight of wood...lbs.	
8.	Weight of water evaporated corrected for moisture in the steam...lbs.	
9.	Weight of ashes and refuse from ash pan... "	
10.	Weight of cinders from smoke box... "	
11.	Percentage of ash, as found by calorimeter test...per cent.	
12.	Total heat of combustion per pound coal as found by calorimeter test...B.t.u.	
	POWER DATA.	
13.	Mean effective pressure, high-pressure cylinders...lbs.	
14.	Mean effective pressure, low-pressure cylinders... "	
15.	Average revolutions per minute...rev.	
16.	Indicated horse power, high-pressure cylinders...H.P.	
17.	Indicated horse power, low-pressure cylinders... "	
18.	Indicated horse power, whole engine... "	
19.	Pull on draw-bar...lbs.	
20.	Dynamometer horse power...H.P.	
	AVERAGES OF OBSERVATIONS.	
21.	Average boiler pressure...lbs.	
22.	Average steam-chest pressure... "	
23.	Average temperature of smoke box...deg.	
24.	Average draft suction...in.	
25.	Average temperature of feed-water...deg.	
26.	Average temperature of atmosphere... "	
27.	Average percentage of moisture in the steam...per cent.	
28.	Maximum percentage of moisture in the steam...per cent.	
	HOURLY QUANTITIES.	
29.	Weight of dry coal burned per hour...lbs.	
30.	Weight of dry coal burned per hour per square foot of grate surface...lbs.	
31.	Weight of coal burned per hour per square foot of heating surface...lbs.	
32.	Weight of water evaporated per hour... "	
33.	Equivalent weight of water evaporated per hour with feed-water at 100° and pressure at 70 lbs...lbs.	
34.	Equivalent weight of water from 100° at 70 lbs. evaporated per square foot of heating surface...lbs.	
	PRINCIPAL RESULTS, COMPLETE ENGINE AND BOILER.	
35.	Coal consumed per I.H.P. per hour...lbs.	
36.	Coal consumed per dynamometer horse power per hour... "	
37.	Weight of "standard coal" consumed per I.H.P. per hour. "	
38.	Weight of "standard coal" consumed per dynamometer horse power per hour...lbs.	

	Whole Run.
BOILER RESULTS.	
39. Water evaporated per pound of coal . . . . lbs.	. . . .
40. Equivalent evaporation per pound of coal from and at 212° "	. . . .
41. Equivalent evaporation per pound of combustible from and at 212° . . . . lbs.	. . . .

CYLINDER DATA.

42. Mean initial pressure above atmosphere . . . . lbs.

	H P Cyl	L P Cyl
43. Cut-off pressure above zero . . . . lbs.	. . . . . .	. . . . .
44. Release pressure above zero . . . . "	. . . . . .	. . . .
45. Compression pressure above zero . . . . . . "	. . . .	. . . .
46. Lowest back pressure above or below atmosphere . . . . lbs.	. . . .	. . . .
47. Proportion of forward stroke completed at cut-off . . . . lbs.	. . . . . .	. . . .
48. Proportion of forward stroke completed at release . . . . lbs.	. . . . .	. . . .
49. Proportion of return stroke uncompleted at compression . . . . lbs.	. .	. . . . . .

CYLINDER RESULTS.

50. Total water consumed per indicated horse power per hour corrected for moisture in steam . . . . lbs.

51. Water consumed per I.H.P. per hour by cylinders alone (from line 51 less all measured losses) . . . . lbs.

	H. P. Cyl.	L. P. Cyl.
52. Steam accounted for by indicators at cut-off lbs.	. . . .	. . . .
53. Steam accounted for by indicators at release. "	. . . .	. . . .
54. Proportion of feed-water used by cylinders (line 52) accounted for at cut-off . . . . lbs.	. . . .	. . . .
55. Proportion of feed-water used by cylinders accounted for at release . . . . lbs.	. . . .	. . . .

Reports should give copy of a set of sample indicator-diagrams, also combined diagram (in case of multi-expansion engine) and a chart showing graphically the principal data.

II. THE ROAD TEST.

A. *Preparation and Location of Instruments.*

The preparations required for the road test, and the proper location of instruments for the purpose, are fully described in Section I, and repetition is unnecessary.

B. *The Dynamometer Car.*

With a suitable dynamometer car the force required to move the train, or the pull upon the draw-bar, is registered upon a strip of paper traveling at a definite rate per mile. The scale upon which this diagram is drawn should be as large as is possible within reasonable limits; a scale of ¼ inch per 1000 lbs. pull being suitable, as the maximum registered pull rarely exceeds 30,000 lbs The height of the diagram should be measured from a base line drawn upon the paper by a stationary pen, so located that when no force is exerted upon the draw-bar the base line should coincide with zero pull.

The apparatus should be arranged to make a record of time marks in connection with the curve showing the pull. A chronometer should be provided, having an electric circuit breaker, by means of which a mark is made on the dynamometer paper every five (5) seconds. A better apparatus may be used in which a continuous speed curve is traced upon the paper, parallel to the curve of pull. The ordinates of this curve, measured from a base line, give the speeds desired.

The location of mile-posts and other points along the route should be fixed upon the dynamometer paper by employing an additional pen, and operating it by means of electric press buttons, which are placed at convenient points in the car.

As already noted, a similar device should be provided for marking upon the dynamometer paper the time of taking indicator diagrams.

The rate of travel of the paper per mile should be such that one inch measured upon the diagrams represents 100 feet for short-distance work, and for long-distance work ½ inch or ¼ inch should be used to represent 100 feet of track. The driving mechanism for the paper should be so arranged that it can be changed to give these three proportions. It is necessary to have all the registering pens located upon the same transverse line at a right angle with the direction of the movement of the paper, in order that simultaneous data may be recorded.

C. *Method of Conducting the Road Test.*

The locomotive having been brought to the train, the steam pressure being at or near the working point, the fire being clean and in good condition though not over 4 to 6 inches thick, the ash-pan being also clean; observations are taken, say five (5) minutes before starting time, of the thickness and condition of the fire, the height of water in the boiler, the depth in the tank, the levels, the water meter, and the air-pump counter; and thereafter the regular observations are carried forward, and coal is fired from the weighed sacks.

Indicator diagrams should be taken as frequently as possible, the intervals between them being not over two minutes.

Other regular observations should be taken at close intervals. Calorimeter readings, when taken, should be continued for at least five (5) minutes at one-minute intervals.

At water stations careful records should be obtained of water heights and levels of boiler and tank.

As the end of the route is approached, the fire should be burned down so as to leave the same amount and the same condition as at the start. When the end is finally reached, the fire should be raked, and its condition carefully noted. If it differs from that which obtained at the beginning, an estimated allowance must be made for such difference.

At the close of the test the height of water in the boiler should be the same as at the beginning, or, if not, the difference, corrected for inclination of the boiler, should be allowed for.

During the process of weighing the coal into the sacks numerous samples should be obtained, and a final sample of these selected. This is to be dried and subjected to chemical analysis and calorimeter test. The sample is weighed before and after drying, and data obtained for determining the weight of dry coal used during the test.

The duration of the road test is the length of time which the throttle-valve is open.

D. The Data and Results.

The data and results of the road test may be tabulated in the form given in Table No. 2. This form corresponds in general with that recommended for the shop test; viz., Table No. 1.

TABLE NO. II.

DATA AND RESULTS OF ROAD TEST ON........ENGINE, MADE........19..

General dimensions, etc. (to be accompanied by a complete description of engine with drawings and dimensions, also of train and route).

1. Kind of engine........ ....
2. Size of cylinders....... .. ..
3. Clearance of cylinders.................... .... per cent.
4. Area of heating surface...... .. .. sq. ft.
5. Area of grate surface.................................. "
6. Size of exhaust nozzles.............. .... ins.
7. Average weight of locomotive and tender (including water) tons.
8. Number of cars..
9. Weight of cars.. tons.

10.	Length of route	miles.
11.	Number of ton miles of train load	ton miles.
12.	Number of ton miles of total load	"
13.	Schedule time of trips	

TOTAL QUANTITIES.

14.	Duration or time throttle-valve is open	hours.
15.	Weight of dry coal burned	lbs.
16.	Weight of water evaporated corrected for moisture in the steam and loss at injector *	"
17.	Weight of ashes and refuse from ash pan	"
18.	Weight of cinders from smoke box	"
19.	Percentage of ash as found by coal calorimeter test	per cent.
20.	Total heat of combustion as found by calorimeter test	B.t.u.
21.	Results of chemical analysis of coal	

POWER DATA.

22.	Mean effective pressure, H.P. cyls	lbs.
23.	Mean effective pressure, L.P. cyls	"
24.	Average revolutions per minute	rev.
25.	Indicated horse power, H.P. cyls	H.P.
26.	Indicated horse power, L.P. cyls	"
27.	Indicated horse power, whole engine	"
28.	Pull on draw-bar	lbs.
29.	Dynamometer horse power	H.P.

AVERAGES OF OBSERVATIONS OF INSTRUMENTS.

30.	Average boiler pressure	lbs.
31.	Average steam-chest pressure	"
32.	Average temperature of smoke box	deg.
33.	Average draft suction	ins.
34.	Average temperature of feed-water	deg.
35.	Average temperature of atmosphere	"
36.	Average percentage of moisture in the steam	per cent.
37.	Maximum percentage of moisture in the steam	"
38.	Weather, wind, etc	

OTHER DATA.

39.	Average position of throttle	
40.	Average position of reversing lever	

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* Should be corrected for steam used by calorimeter, air pump, blower, safety valve, and whistle, to find cylinder results — line 56.

41. Average speed in miles per hour......................
42. Maximum speed in miles per hour....................
43. Number of stops..................................
44. Average number of strokes of air pump per minute.......
45. Total estimated weight of steam used by air pump per hour lbs.
46. Estimated loss of steam at safety valve per hour........ "
47. Estimated loss of steam at whistle per hour............ "
48. Estimated weight of steam used by blower per hour..... "
49. Estimated loss of steam at calorimeter per hour......... lbs.

HOURLY QUANTITIES.

50. Weight of dry coal burned per hour..... .... lbs.
51. Weight of dry coal burned per hour per square foot of grate surface... .... "
52. Weight of coal burned per square foot of heating surface. "
53. Weight of water evaporated per hour. . . .. "
54. Equivalent weight of water evaporated per hour with feed-water at 100° and pressure 70 lbs .... "
55. Equivalent weight of water from 100° at 70 lbs. evaporated per square foot of heating surface.... .. .. "
56. Weight of water consumed by engine cylinders (line 53, less sum of lines 45, 46, 47, 48, and 49).. .... "

PRINCIPAL RESULTS — COMPLETE ENGINE AND BOILER.

57. Coal consumed per I.H.P. per hour.................... lbs.
58. Coal consumed per dynamometer horse power per hour... "
59. Coal consumed per ton mile of train load... .... "
60. Coal consumed per ton mile of total load............... "
61. Weight of standard coal consumed per I.H.P. per hour. . "
62. Weight of standard coal consumed per dynamometer horse power per hour..... .... "
63. Weight of standard coal consumed per ton mile of train load.................... .... "
64. Weight of standard coal consumed per ton mile of total load "

BOILER RESULTS.

65. Water evaporated per pound of coal.................. lbs
66. Equivalent evaporation per pound of coal from and at 212° "
67. Equivalent evaporation per pound of combustible from and at 212°..................................... "

CYLINDER DATA.

68. Mean initial pressure above atmosphere................ lbs.

	H P. Cyl.	L. P. Cyl.
69. Cut-off pressure above zero........... ....lbs.	......	......
70. Release pressure above zero..................... "	....	....
71. Compression pressure above zero............. ... "	....	......
72. Lowest back pressure above or below atmosphere.. "	....	....
73. Proportion of forward stroke completed at cut-off....	....	....
74. Proportion of forward stroke completed at release....	....	....
75. Proportion of return stroke uncompleted at compression	. ....	... .
76. Mean effective pressure (lines 22 and 23) . ..... lbs.	. .....	. ...

CYLINDER RESULTS.

77. Total water consumed per indicated horse power per hour corrected for moisture in steam.... . . .. lbs.

78. Water consumed per I.H.P. per hour by cylinders alone (from line 56)... .. .. "

	H. P Cyl	L. P Cyl.
79. Steam accounted for by indicator at cut-off... ..lbs.	. ....	....
80. Steam accounted for by indicator at release. ... "	....	....
81. Proportion of feed water used by cylinders (line 78) accounted for at cut-off...	. . ....	......
82. Proportion of feed water used by cylinders accounted for at release..	......	....

365. Calorimetric Methods of Engine Testing. — The object of calorimetric tests is the determination of the various ways in which the heat supplied to the engine is distributed at various points in a single cycle, to bring out the heat interchanges between steam and cylinder walls, etc. The methods of carrying on this work were mainly developed by Hirn, and hence the computation is often called *Hirn's Analysis*. It is also often called simply the *heat analysis*, especially if modifications of Hirn's original methods are here and there introduced in the computation.

The method consists of dividing the average indicator card obtained on a complete engine test into its several events and for each event computing the quantities of heat involved, that is, the quantity of heat originally available at the beginning, the quantity remaining at the end, the quantity converted into work, and the quantities that can be accounted for as lost to cylinder walls by radiation, etc. A very complete exposition of the entire method of computation, together with a concrete example of its application, will be found in Peabody's "Thermodynamics of the Steam Engine," to which the student is referred. The following is an abstract of the scheme there developed.

Let Fig. 503 represent the average indicator card. Draw in the line of zero pressure and the line of zero volume, V_c being the volume of clearance. Number the important events on the card as follows:

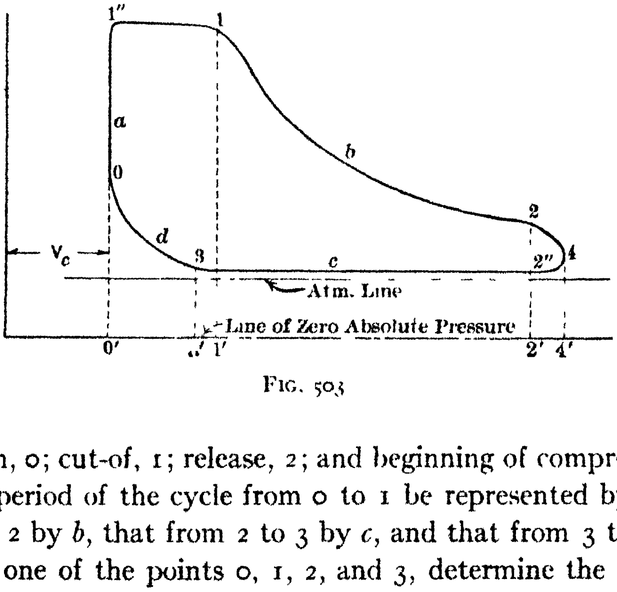

FIG. 503

Admission, 0; cut-of, 1; release, 2; and beginning of compression, 3. Let that period of the cycle from 0 to 1 be represented by a, that from 1 to 2 by b, that from 2 to 3 by c, and that from 3 to 0 by d. For each one of the points 0, 1, 2, and 3, determine the following data: Absolute pressure; heat of liquid, q; internal latent heat, ρ; heat of vaporization, r; total heat, λ; and specific volume of steam, v. Also measure the total volume (= clearance volume + volume from end of stroke to point) up to the various points. Record all these data on the following blank form. If at any of the points the steam is superheated, several of the quantities above are correspondingly modified.

ABSOLUTE STEAM PRESSURES FROM INDICATOR CARDS AND CORRESPONDING PROPERTIES OF STEAM.

CYLINDER {High Pressure or Low Pressure.

	Cut-off.	Release.	Beginning of		Symbol.
			Compression.	Admission.	
Subscript used	1	2	3	0	
Absolute steam pressure . lbs.					
Heat of the liquid . B.t.u.					q
Internal energy "					ρ
Latent heat of evap. "					r
Total heat in dry and saturated steam . B.t.u.*					H or λ
Specific volume of steam, dry and saturated . cu. ft.*					v
Volume symbols	$V_c+V_1=$	$V_c+V_2=$	$V_c+V_3=$	$V_c+V_0=$	$V_c=$ clearance vol. =
Volumes at the various points, as measured from card cu. ft.					

* Where steam is superheated, modify these items to suit.

MEAN PRESSURES AND HEAT EQUIVALENTS OF THE EXTERNAL WORK.

CYLINDER {High Pressure or Low Pressure

	Subscript	Mean Pressures.	External Work	External Work.
			Foot-Pounds	B.t.u.
Symbol used.		M.E.P.	W	AW *
Admission	a			
Expansion	b			
Exhaust	c			
Compression	d			

* $A = \frac{1}{778}$.

Next divide the card by the broken lines shown into areas representing the external work done during the various events of the cycle. Thus:

External work done during admission = area 0′, 1″, 1, 1′.
" " " " expansion = area 1, 2, 2′, 1′.
" " " " release and exhaust = area 2, 4, 4′, 2′ minus area 3, 4, 4′, 3′.
" " " " compression = area 0, 3, 3′, 0′.

For each one of these areas find the mean ordinate by means of the planimeter and compute the M.E.P. by proper spring scale. Record these on the form where indicated. For each event compute and record next the foot-pounds ($= W$) of external work done. This is equal to

$W =$ M.E.P. × piston area in square inches × distance moved by piston during the event.

Note. — During release and exhaust the piston moves from 2 to 4 in one direction and from 4 to 3 in the opposite direction. Since the M.E.P. recorded for this event is the net, as above computed, the distance moved by the piston in this event, for the computation of W, should be taken simply as 4 to 3.

Convert the various quantities W_a, W_b, W_c, and W_d of external work in foot-pounds into the heat equivalents AW_a, etc., by dividing each by 778, and record on the form.

From the engine test determine the actual weight of cylinder steam furnished to the engine per cycle. This does *not* take into account the weight of jacket steam, if jackets are used, as the action of this jacket steam can only modify the heat interchange between walls and cylinder steam, and this action will appear in the analysis. Call this weight of cylinder steam $= M$ pounds. Compute the weight of steam caught in the clearance spaces per stroke, on the assumption that the steam is dry at point 3, and call this weight $= M_0$ pounds.

This completes the computation of preliminary data. Before proceeding to establish energy balances for the various points on the card, we must know the quality of the steam at points 0, 1, 2, and 3. The quality at point 3 is usually assumed $= 1.0$. The quality of the other points can be computed as follows:

The specific volume of one pound of mixed vapor and liquid is

$$v = xu + \sigma,$$

where

$u =$ volume increase during vaporization,

$\sigma =$ specific volume of water under the temperature conditions existing.

The volume V at any point on the indicator card may then in general be expressed by the equation

$$V = Mv = M(xu + \sigma).$$

Hence for the points 0, 1, and 2, we have

$$V_c + V_0 = M_0\,(x_0 u_0 + \sigma_0)$$
$$V_c + V_1 = (M + M_0)\,(x_1 u_1 + \sigma_1)$$
$$V_c + V_2 = (M + M_0)\,(x_2 u_2 + \sigma_2).$$

In these equations all the factors but x_0, x_1, and x_2 are known, and these may therefore be found. The value of σ changes but very little (from .016 to .018) in the ordinary pressure range, and is in any case so small that it is generally neglected.

Note. — Where a saturation curve has already been constructed for the general engine test, the qualities of the steam at points 1 and 2 are obtained more quickly by means of this curve.

The heat, or rather the energy, balances may next be established for the various points on the diagram as per the following scheme. This is carried through, assuming that at all the points considered the steam is saturated, wet or dry, but is at no time superheated. If superheat occurs at any point the result for x in some of the equations above will be greater than unity. In that case, the computations are modified by substituting, in the proper places, for the intrinsic energy (all the equations in which ρ occurs) and for the total energy (all the equations in which r occurs), the following expressions:

$$H_{\text{intrinsic}} = \lambda + C_p\,(t_1 - t) - \text{total } APu \text{ value},$$
$$H_{\text{total}} = \lambda + C_p\,(t_1 - t),$$

where t_1 and t are the temperatures of the superheated steam and of saturation respectively. t_1 is computed from the computed value of x according to the equation

$$\frac{r\,(x - 1)}{C_p} + t = t_1.$$

Here C_p = mean specific heat for the range $t_1 - t$. Hence this equation will have to be solved by trial.*

Energy balance at cut-off (*Point* 1).

(*a*) Intrinsic energy in steam at admission,

$$H_0 = M_0\,(x_0 \rho_0 + q_0).$$

(*b*) Total energy added from 0 to 1 (Period *a*),

$$H = M\,(xr + q)$$

Here x is the quality of steam in the steam pipe at the throttle.

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* The use of the Mollier diagram, Fig. 245, will avoid many of these computations.

If the steam is superheated at this point, use the equation given above for the total energy in superheated steam. Note the use of ρ instead of r in the equation for (a). The APu value is furnished by the boiler.

(c) Energy utilized in external work (Period a)

$$= AW_a.$$

(d) Intrinsic energy in steam at cut-off,

$$H_1 = (M + M_0)(x_1\rho_1 + q_1).$$

(e) Heat interchange between steam and cylinder walls (Period a),

$$Q_a = H + H_0 - H_1 - AW_a.$$

Energy Balance at Release (Point 2).—

(a) Intrinsic energy in steam at cut-off,

$$H_1 = (M + M_0)(x_1\rho_1 + q_1).$$

(b) Energy utilized in external work (Period b)

$$= AW_b.$$

(c) Intrinsic energy in steam at release (Point 2),

$$H_2 = (M + M_0)(x_2\rho_2 + q_2).$$

(d) Heat interchange between steam and cylinder walls (Period b),

$$Q_b = H_1 - H_2 - AW_b.$$

Energy Balance at Beginning of Compression (Point 3).—

(a) Intrinsic energy in steam at release,

$$H_2 = (M + M_0)(x_2\rho_2 + q_2).$$

(b) Energy equivalent of work done by piston upon steam

$$= AW_c.$$

(c) Total energy rejected in exhaust,

$$H_{ex.} = M(x_{ex.}r_{ex.} + q_{ex.}).$$

(d) Intrinsic energy in steam at beginning of compression (Point 3),

$$H_3 = M_0(x_3\rho_3 + q_3).$$

(e) Heat interchange between steam and walls (Period c),

$$Q_c = H_2 - H_3 - M(x_{ex.}r_{ex.} + q_{ex.}) + AW_c.$$

The total energy rejected (Item (*c*) above), if the engine is condensing, may also be found as follows:

Let the weight of condensing water used per M pounds of steam be G pounds and let the heat of the liquid of the inlet water be q_i, that of the outlet water q_k. Also let the heat of the liquid remaining in the steam condensed be q_5. Then

$$\text{Heat in condensing water} = G(q_k - q_i),$$
$$\text{Heat in condensed steam} = Mq_5.$$

The sum of these two quantities is the heat rejected. This sum will, in general, not check with the quantity computed under (*c*) above on account of radiation losses from condenser and piping. It is, however, rather difficult to determine the quality of steam when the latter is under a vacuum, as it is likely to be in the exhaust of a condensing engine. Consequently, this method of computing Item (*c*) is mostly used. It should be remembered that in this case the quantity Q_c includes the radiation from condenser and piping.

Energy Balance at Admission (Point o*).*—

(*a*) Intrinsic energy in steam at beginning of compression,

$$H_3 = M_0\,(x_3\rho_3 + q_3).$$

(*b*) Energy equivalent of work done by piston upon steam (Period *d*)

$$= AW_d.$$

(*c*) Intrinsic energy in steam at admission (Point o),

$$H_0 = M_0\,(x_0\rho_0 + q_0).$$

(*d*) Heat interchange between steam and walls,

$$Q_d = H_3 - H_0 + AW_d.$$

Net Heat Interchange between Steam and Cylinder Walls.—

$$Q_{net} = Q_a + Q_b + Q_c + Q_d.$$

Radiation Loss from an Engine.—

Let Q_r = radiation loss.
H = heat supplied in cylinder steam per cycle.
H_j = heat supplied in jacket steam per cycle.
H_j' = heat rejected in jacket steam per cycle.

Then in connection with the other equations developed above for heat rejected from the cylinder and for heat transformed into work, we may write:

$$Q_r = H + H_j - H_j' - G(q_k - q_1) - Mq_b - A(W_a + W_b - W_c - W_d).$$

The heat supplied by the steam jacket per cycle, if M_j pounds of jacket steam are condensed, equals

$$H_j = M_j(x'r' + q'),$$

where x', r', and q' refer to the condition of the jacket steam as it enters the jacket. Should this steam be superheated at entrance, the equation becomes

$$H_j = r' + q' + C_p(t_j' - t'),$$

where t_j' = temperature of steam and t' = saturation temperature. The heat discharged by the jacket is in every case

$$H_j' = M_j q'',$$

where q'' = heat of the liquid of the discharged jacket condensation. In most cases q'' is very nearly equal to q'.

In case no jacket is used, H_j and H_j' of course become 0. It should be noted that the quantity Q_r is determined by difference, but Q_r is small as compared with the other quantities, and any error in the latter is consequently likely to be a considerable percentage of Q_r, for which reason the determination of radiation by this method is somewhat uncertain.

Heat Analysis Applied to Multicylinder Engines. — The method of computation for each cylinder is the same as outlined above. To determine the heat quantity flowing from one cylinder to the next, means of finding the quality of the steam must be provided. When the steam is under a vacuum a determination of quality may sometimes be made by connecting the outlet of the calorimeter to the condenser.* Where it is not possible actually to find the quality between cylinders, an approximation to the heat supplied each cylinder may be made by determining the radiation for the whole engine as indicated in the equation above for Q_r, except that AW will represent the work done in all the cylinders. Divide the total radiation loss by the number of cylinders, assuming that the loss is the same from each cylinder. Having the radiation loss from any given cylinder determines the heat discharged in exhaust by equation,

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* On account of the pulsating flow in the piping between cylinders, the quality is likely to change from the beginning of an exhaust stroke to the end of the same stroke. It is not certain that calorimeters of the type ordinarily employed indicate true average quality under these conditions.

$$\text{Heat discharged} = \text{heat supplied} - \text{heat transformed into work} - \text{heat lost by radiation.}$$

It is needless to say that this method may introduce errors sufficiently great to make the entire work useless and that the heat analysis can be applied to a multicylinder engine with a fair degree of accuracy only if quality determinations of the steam in the exhaust of all the cylinders are made.

366. Valve Setting on Steam Engines. — This work is not strictly of an experimental engineering nature, but is introduced in many laboratory courses for the purpose of familiarizing the student with the general procedure of setting valves. No specific directions for this work can be given here on account of the multiplicity of different valve gears, and in every case it will therefore become necessary that directions be written specifically for the engine or engines that may be in possession of any given laboratory.

The student is referred to books on valve gears. See "The Slide Valve," Begtrup; "Handbook of the Corliss Steam Engine," Shillitto; "Valve Setting," Collins; "Slide Valve Gears," Halsey; "Valve Gears," Spangler.

CHAPTER XIX.

STEAM TURBINES.

367. General Considerations. — The present commercial importance of the steam turbine makes necessary an analysis of turbine efficiencies and turbine losses. It is not possible, in the scope of this book, to enter into a discussion of the details of turbine design, even in so far as they may affect the efficiency developed, especially since the *method of testing*, which is the main topic of this chapter, is practically the same for all types of turbines. For that reason, only, so much of the differences in construction of the various types as is sufficient to make clear the meaning of some technical terms generally used in connection with classification will be presented.* At the same time the main features of the important commercial types will be briefly illustrated.

368. Types of Turbines. — Turbines are generally divided into two classes: the *impulse type* and the *reaction type*. Moyer states that without further explanation the use of these terms might be very misleading, as practically all commercial types of turbine today operate both by impulse and by reaction. The same author clearly defines what is meant by the terms as actually applied. Figures 504, 505, and 506 are reproduced from his work. In each case A represents an expanding nozzle in which steam is expanded from boiler pressure to some lower pressure, thereby acquiring high velocity and kinetic energy. In Fig. 504 the blades have single curvature, and the steam flows through the blade passage B without being turned back upon itself to any degree. If the wheel were held stationary, the steam would escape from the blade passages practically parallel to the shaft axis. The only force that the steam can exert upon the wheel is impulse, and the construction shown typifies a pure impulse turbine. If, as in Fig. 505, the blades have double curvature, although the blade passage *B* is otherwise the same, the steam will be partly turned back upon itself, a reaction force will be produced, and both impulse and reaction will therefore be at work. The type of wheel that in practice would be called the reaction type is shown in Fig. 506. Note first the change in the blade form as compared with the two previous forms. But the main change lies in the fact that the blade passage *B* allows of the expansion of the steam while passing through it. The nozzle *A* is so designed that only a part of the expansion occurs in it, the remainder down to back pressure occurring in the passage *B*.

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* For full discussion of turbine details see any of the following books: Stodola, "The Steam Turbine"; Moyer, "Steam Turbines"; French, "Steam Turbines"; Jude, "The Theory of the Steam Turbine"; Neilson, "The Steam Turbine"; Thomas, "The Steam Turbine."

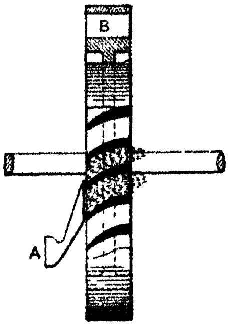

FIG. 504.

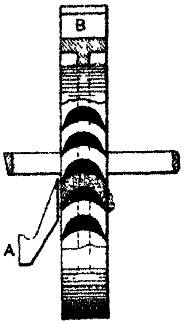

FIG. 505.

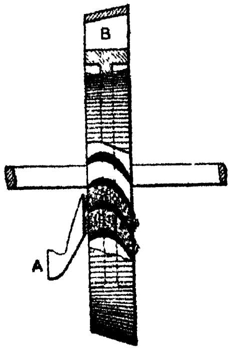

FIG. 506.

The main distinction in practice between impulse and reaction turbines apparently lies in the fact that in the first type no expansion of the steam takes place in the blade passages, while in the latter type it does take place. If two pressure gauges were connected to the entrance or exit sides of a wheel, we would therefore have an impulse turbine if the gauges showed the same pressure, and a reaction turbine if the gauge on the exit side showed a pressure less than the other. Under that definition the wheel shown in Fig. 505 is an impulse turbine, that shown in Fig. 506, a reaction turbine, but note that both of these wheels really operate by impulse and reaction combined. The following four classes of turbines employing the principles above outlined, practically include all commercial turbines.

369. The Single-stage Impulse Turbine. — In this turbine the wheel is of the type of Fig. 505. The steam expands in the nozzle *A*, Figs. 507 *a* and *b*, down to back pressure, increasing the velocity to the maximum. Passing through the blade passage *B* reduces the velocity according to the work performed. The curves of Figs. 507*c* and *d* clearly show the pressure and velocity changes occurring in the steam.

The commercial example of this turbine is the De Laval, Figs. 508 and 509. In the latter figure, *A* is the turbine wheel. Owing to the fact, first, that the velocity of the steam leaving the nozzle is very high in this turbine, on account of the single expansion, and second, that the best efficiency of the turbine is realized when the peripheral speed of the wheel is about 50 per cent that of the steam, the peripheral speed of these turbines is quite high (about 1350 feet per second). This makes it necessary in general to employ speed-reducing devices before applying the power developed to commercial machines. This gear for one of the smaller turbines is indicated in Fig. 509 at *J*, *K*, *L*. *K′* is an electric generator, coupled at *M*. In the larger sizes the pinion *J* is located between two gears, so that the turbine drives two shafts.

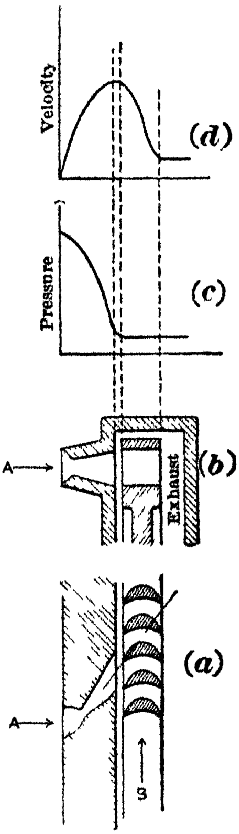

FIG. 507.— SINGLE-STAGE IMPULSE TURBINE.

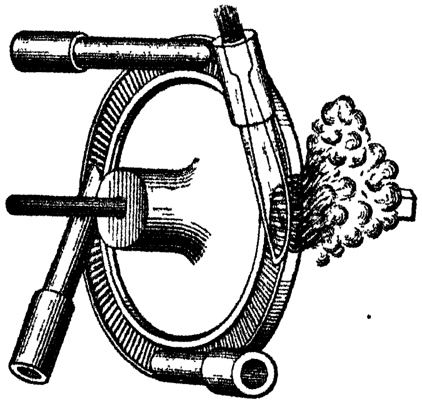

FIG. 508. — WHEEL AND NOZZLES OF DE LAVAL TURBINE.

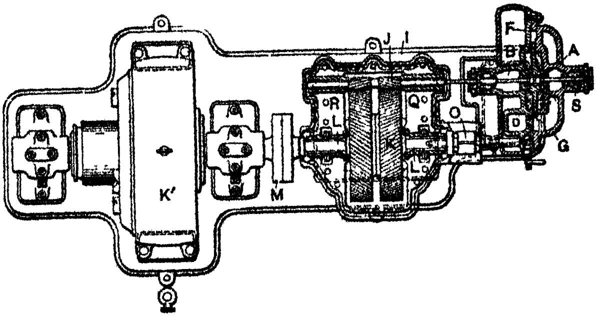

FIG. 509.—DE LAVAL STEAM TURBINE.

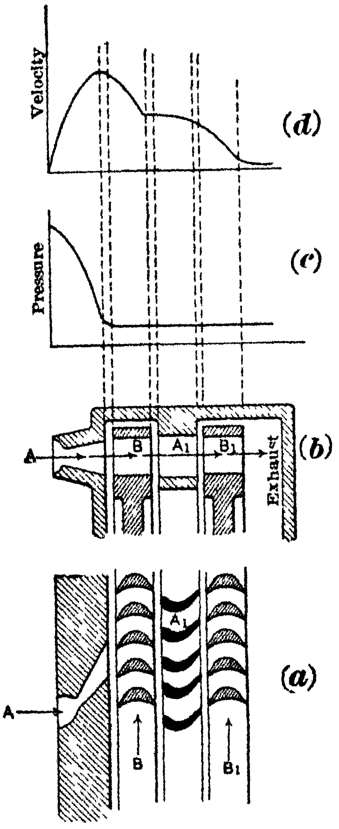

FIG. 510.—IMPULSE TURBINE WITH TWO VELOCITY STAGES

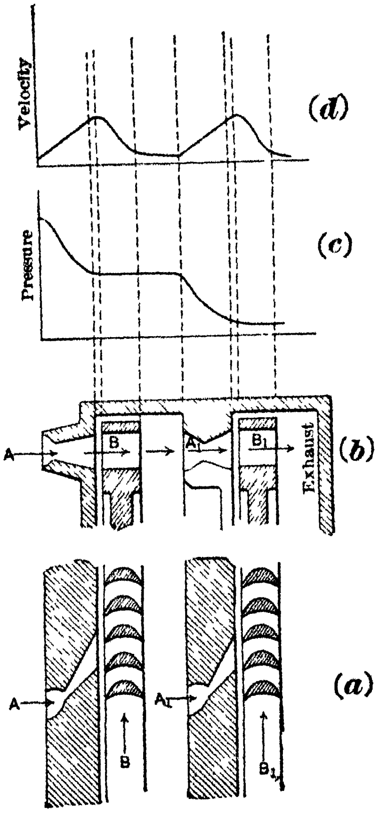

FIG. 511.—IMPULSE TURBINE WITH TWO PRESSURE STAGES.

370. The Multi-stage (Compound) Impulse Turbine. — Two kinds of this type of turbine are recognized, according to whether the machine has several velocity or several pressure stages. Some commercial machines have both kinds of stages, hence the name compound.

Figures 510*a* to *d* explain the operation of a turbine having two *velocity stages*. The operation of the first nozzle *A* and moving passage *B* is the same as for the single-stage turbine just described. Note that the pressure and velocity changes are also the same. After leaving the passage *B*, the steam next enters the second set of stationary nozzles A_1. These are really nothing but guide blades to cause the steam to strike the second set of movable blade passages B_1 at the proper angle. The velocity curve shows that the velocity changed in *B*, owing to work done, but that little change took place in A_1, except that due to friction. The second movable set B_1 then reduces the velocity to the minimum. The advantage of this construction over that of the single-stage turbine is that the peripheral speed may be kept down. Three- or four-stage turbines of this type have been built, but they are not common.

Figures 511*a* to *d* show the operation of a turbine having two *pressure stages*. In this construction the first-stage stationary nozzles *A*, followed by the movable set of blades *B*, are practically the same as Fig. 507, the single impulse wheel. The same statement holds for the second stage, except, of course, that the pressure on the second nozzle, A_1, is the same as the exit pressure from the first movable set of blades. Note the pressure and velocity changes in Figs. 511*c* and *d*, as compared with those in Figs. 510*c* and *d*.

The purely velocity-stage type of turbine is apparently not much used commercially. Machines using the pure pressure-stage principle are the Rateau, the Zoelly, and the Hamilton-Holzworth. The pressure and velocity changes, together with the blade and guide construction, are well shown in Fig. 512.* A turbine compounding velocity and pressure stages is the Curtis, made by the General Electric Company. This machine in the larger sizes is of the vertical type. In Fig. 513,† *C* is the turbine, *B* the electric generator, and *D* the step bearing upon which the construction rests. *A* is a centrifugal governor which controls the speed through the medium of an hydraulic cylinder, which in turn controls the opening or closing of poppet valves by which steam is admitted to the various nozzles in the first stage. Figure 514* shows two pressure stages from one of these turbines. The first set of nozzles *A* reduces the pressure and increases the velocity. This set of nozzles is followed in the same stage by sets of movable blades and of guide blades, in this case three of the former and two of the latter. In passing through the movable blades the velocity is reduced, while in passing through the guide blades it is kept practically constant. On leaving the last set of movable blades in this stage, the pressure is practically the same as it was after leaving the nozzles *A*. The steam next encounters a second set of stationary nozzles A_1 in which the pres-

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* French, "Steam Turbines," p. 109.

† French, "Steam Turbines," p. 117.

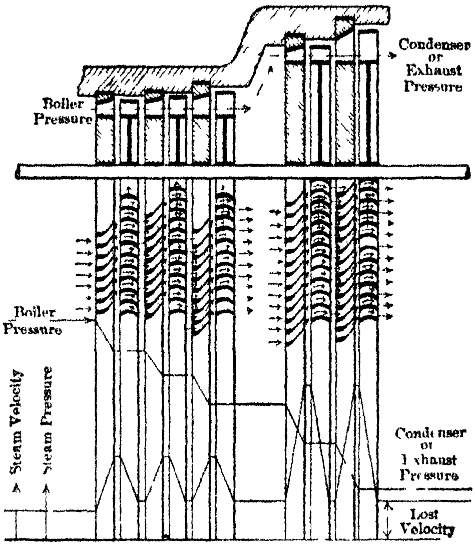

FIG. 512.—MULTI-STAGE TURBINE (FIVE PRESSURE STAGES).

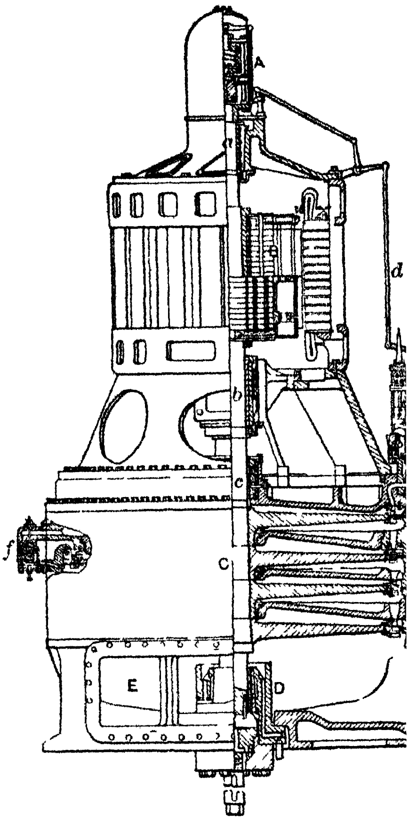

FIG. 513.—CURTIS TURBINE.

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* French "Steam Turbines," p. 114.

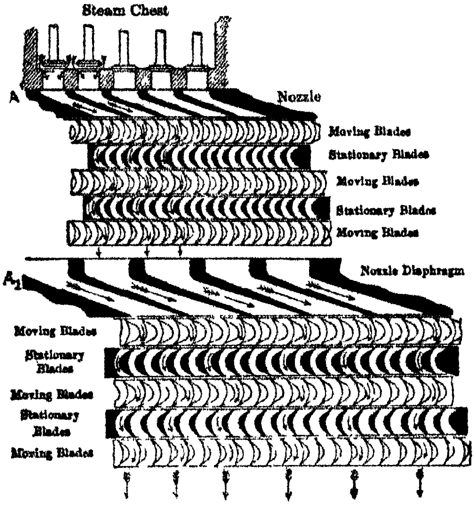

FIG. 514.—TWO PRESSURE STAGES OF A CURTIS TURBINE

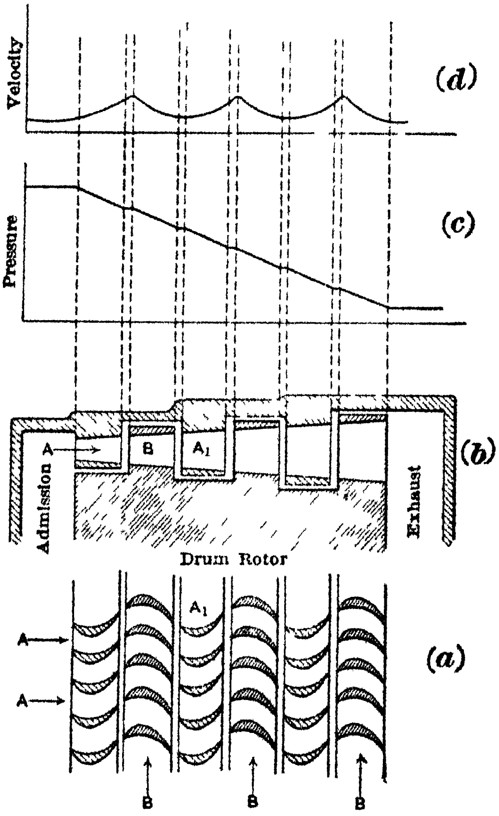

FIG. 515.—MULTI-STAGE REACTION TURBINE.

sure is again reduced with a gain in velocity, and the second velocity stage then follows.

371. The Multi-stage Reaction Turbine.—See Figs. 515*a* to *d*, concerning the principles of operation. There are no special nozzles, the steam from the steam chest entering at once the first set of stationary guide blades *A*. In these, expansions take place and the pressure drops a certain amount with a gain in velocity (see Figs. 515*c* and *d*). In the movable set of blades *B* further expansion takes place and the pressure drops, but owing to the work done the net result is a loss of velocity. In the second set of guide blades A_1, the velocity is again increased by a further pressure drop, and thus the velocity is alternately increased in the stationary blades and decreased in the movable blades, by pressure drop in each case, until exhaust pressure is reached.

One of the most important commercial examples of this type of turbine is the Parsons, built by the Westinghouse Machine Co.

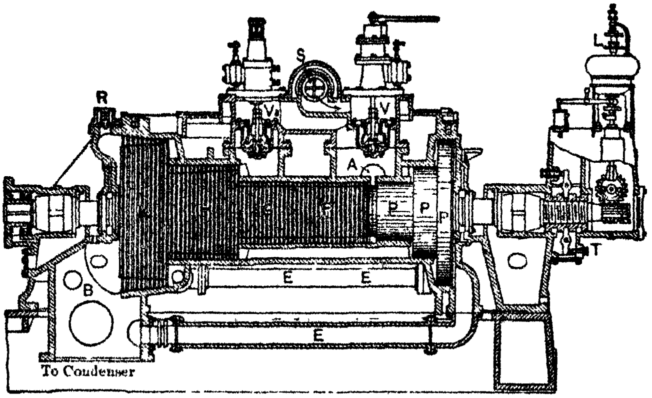

FIG. 516 —WESTINGHOUSE-PARSONS STEAM TURBINE.

Fig. 516 shows the construction of one of these machines. Another type built by the same company is a double-flow turbine designed to avoid the end thrust present in the ordinary construction. This machine is, however, really compounded, in that it has a set of nozzles with two velocity stages, one intermediate section, and two low-pressure sections of Parsons blading.*

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* Moyer, "The Steam Turbine," p. 173.

The Allis-Chalmers turbine is of the Parsons type, the main differences being in mechanical details of construction.

372. Turbines of the Pelton Type (Open-vane or Bucket Turbines). — Turbines of this type were developed by Rateau in Paris and by Riedler and Stumpf in Berlin. Riedler-Stumpf turbines of considerable size were built and are in operation. In this country these turbines are built by Sturtevant, by Kerr, by Terry, and others. The general construction of the wheels used is very well shown in Fig. 517,* which illustrates a Sturtevant wheel. The similarity to the ordinary Pelton water-wheel is striking. Figure 518 † shows a section of a Kerr turbine having five compartments and wheels. The position of the nozzles with reference to the wheel is well shown in Fig. 519.†

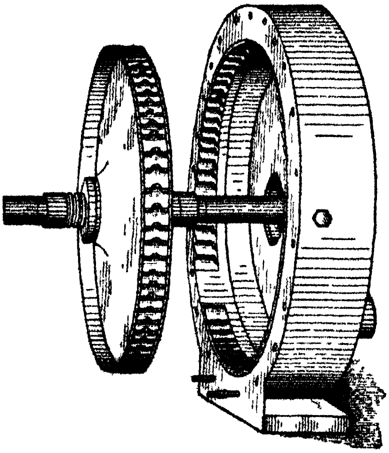

FIG. 517. — WHEEL AND CASING OF STURTEVANT TURBINE.

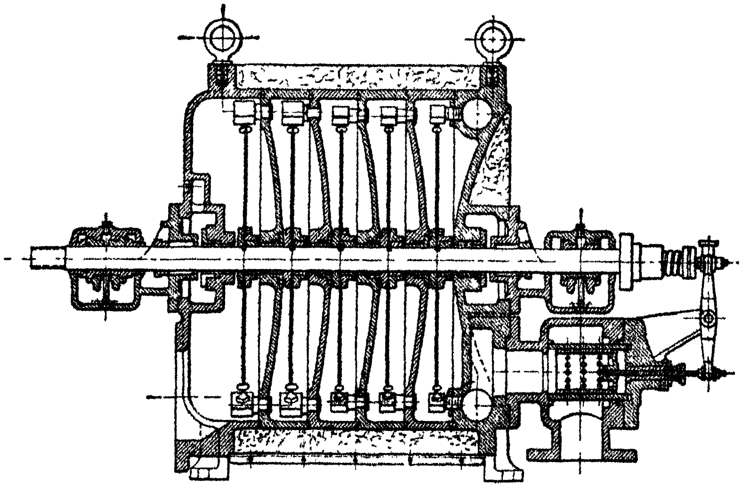

FIG. 518. — KERR TURBINE.

* Moyer, "Steam Turbine," p. 208.
† Ibid., p. 213.

373. **The Testing of Steam Turbines.** — Commercial testing of steam turbines is a comparatively simple matter. There is no direct way of determining the actual work done by the steam in the rotating parts of the turbine; that is, there is no test quantity analogous to the "Indicated Horse Power" of reciprocating engines. For an economy, efficiency, or capacity test, therefore, the measurements made are the amount of steam used, the condition of the steam at inlet (and in exhaust if possible), and the power output.

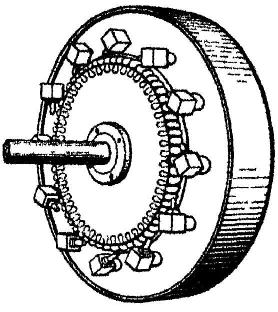
FIG. 519.—WHEEL AND NOZZLE OF KERR TURBINE.

The arrangements for the measurement of steam used are the same as for the reciprocating engines. For the purpose of obtaining the best economy, the great majority of turbines are operated condensing,* and the usual type of condenser is the jet. This means that in most cases measurement of feed water to the boilers will have to be resorted to.

Depending upon whether the turbine operates upon superheated or saturated steam, either a thermometer is inserted or a calorimeter is attached to the steam pipe near the main throttle. For the precautions necessary to observe, see Chap. XIV. It has been found that the effect of superheated steam upon economy is more marked in the case of turbines than in that of reciprocating engines,† and many turbines operate on superheated steam. That simplifies the determination of steam quality. Concerning the determination of quality of steam in exhaust, this is desirable as a matter of scientific data. If the steam was originally sufficiently superheated, it may happen that the steam is still superheated in exhaust, in which case the quality determination consists merely in determining the pressure and temperature. Where this is not the case, the determination of quality in exhaust becomes difficult in condensing turbines. The ordinary calorimeter is practically useless on account of the difficulty of inducing flow. The shunting of a part of the exhaust steam through an electric calorimeter, of the type of the Thomas, for instance (see Chap. XIV), may solve the problem, but while the use of such an instrument is possible in a laboratory, it is generally out of question in testing in the field.

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* For discussions of the effect of different vacuums upon economy, see any of the works above mentioned.

† Mainly on account of decreased friction due to elimination of water particles in the steam. See works mentioned.

DEPARTMENT OF EXPERIMENTAL ENGINEERING, SIBLEY COLLEGE.

REPORT OF STEAM-TURBINE TEST.

Made by.................................... *Date*..................

....*Turbine Mfg. by*....................

Run No.	I.	II.	III.	IV	V.
1. Duration of test.					
2. Barometer, in. Hg					
3. Speed, R.P.M.					
4. Steam pressure, lbs. gauge					
5. " " lbs. abs.					
6. Exhaust pressure, in. vacuum					
7. " " lbs. gauge					
8. " " lbs. abs.					
9. Superheat of steam, deg. Fahr.					
10. Quality of steam, per cent dry.					
11. Condensed steam per hour, lbs.					
12. Volts					
13. Amperes, line					
14. " field					
15. " total					
16. Power output of generator, K.W.					
17. Efficiency of generator, per cent					
18. Generator losses, K.W.					
19. Power input to generator, K.W., or B.H.P.					
20 Power output of steam turbine, internal H.P.					
21. Steam consumption per K.W.-hr. Set					
22. " " " internal H.P.-hour					
23. B.t.u. supplied per K.W.-hr. Set					
24. " " " internal H.P.-hour					
25. B.t.u. available per lb. steam. Clausius cycle					
26. Theoretical water rate. Clausius cycle					
27. B.t.u. utilized per lb. steam. Set					
28. " " " " Internal H.P.					
29. Potential efficiency, over all					
30. " " based on internal H.P.					
31. Temp. injection water, deg. Fahr.					
32. Temp. discharge water, " "					
33. Temp. condensed steam, " "					
34. Condensing water per hr., lbs.					
HEAT BALANCE.					
35. Heat supplied, B.t.u.					
36. " utilized "					
37. " discharged "					
38. " lost by radiation, leakage, etc., B.t.u.					

The question of determining the power output of turbines is generally easily solved, because most of them are used for the generation of electric current. The switchboard readings, therefore, determine the output. This corrected for generator efficiency, which can be separately determined or computed but is best obtained from the builders of the machine, gives the actual mechanical work delivered by the turbine to the generator.

It occasionally happens that the power developed is used in some other way. In such a case a fluid friction brake (see Chap. X) is best on account of the high speed, especially if the power to be absorbed is considerable. The power developed by marine steam turbines is best measured by means of some type of torsion meter.

The preceding form is convenient for recording the data from a turbine test and also points out the principal results to be computed.

374. Computations in Connection with Steam-turbine Tests. — If the turbine is tested for economy at a series of loads, Willan's line (see p. 726) should be plotted during the tests as a check upon the work, as in the case of a reciprocating engine. The final report should show economy curves drawn between steam consumption per horse power and kilowatt-hour and output (see Chap. XVIII).

Internal Horse Power. — Where the turbine is driving a generator, the output will be determined in kilowatts. This is transposed to *electrical horse power* by multiplying by $\frac{1000}{746} = 1.34$. Dividing the electrical horse power next by the efficiency of the generator gives what is the equivalent of *brake* or *shaft horse power* in a reciprocating steam engine. It has already been pointed out that there is no indicated horse power in the case of steam turbines, but in order to make the results of steam turbine tests comparable with those of reciprocating engines, the term *internal horse power* is invented. This is obtained from the brake horse power by taking into account the mechanical efficiency of a reciprocating engine of the same capacity and operated at the same load. The assumption of this efficiency is, of course, largely a matter of estimation based on the results of available tests. For instance, in a test quoted by Moyer, the brake horse power of a 400-K.W. turbine was found to be 660. From available data it was assumed that the mechanical efficiency of a reciprocating engine would be 93.3 per cent when operated at the same load and under the same conditions. The internal horse power, therefore, is $\frac{660}{.933} = 708$.*

Thermal Efficiency. — As in the case of reciprocating engines, this is defined as the ratio of the heat equivalent of the work done in a given time divided by the heat supplied the turbine in the same time. For the mathematical expression of this ratio, see Chap. XVIII.

B.H.P.	594.9	Heat in dry and saturated steam, per lb.	1195.4	B.t.u.
Internal H.P. (efficiency = .94)	632.5	Heat in superheat, per lb.	62.0	"
Total steam per hour, lbs.	7384	Total heat per lb. of steam above 32° F.	1257.4	"
Steam per internal H.P. per hour, lbs.	11.67	Heat in condensed steam	52.3	"
Absolute steam pressure, lbs.	170.0	Heat in steam per lb. above temp. of condensed steam	1205.1	"
Superheat, degrees	109.0	Thermal efficiency based on total heat in steam above 32° $= \frac{2545}{11.67 \times 1257.4} =$	17.3%	
Temp. condensed steam	95.8	Thermal efficiency based on total heat in steam above temp. of condensed steam $= \frac{2545}{11.67 \times 1205.1} =$	18.1%	

There appears to be some difference in practice as far as computing the heat supplied in a given time is concerned. In all cases the total heat in the steam supplied is computed above 32 degrees. Most authorities subtract from this the heat of the liquid left in the condensed steam. This introduces a factor into the computation which is peculiar to the individual plant, depending upon the operation of the condensing plant, irrespective of the pressure in the exhaust. It is hard to see why, in the computation of a criterion by which the performance of the turbine itself is to be judged, any factors independent of the turbine should be considered. It is true, of course, that this matter is largely one of definition, and, as stated above, the subtraction of the heat of the liquid in the condensed steam is usually made. To show the difference that occurs in the results when this allowance is made and when it is not made, the figures on p. 820 are quoted from French*:

---

* It may be stated that "internal horse power" is based upon arbitrary definition, and that for the purpose of getting an equitable basis upon which to base scientific comparisons it is much better to express the steam or heat consumption of either engines or turbines upon the brake horse power.

Potential (Clausius) or Cylinder Efficiency. — The theoretical cycle of the steam turbine is the Clausius, in which there is assumed to be adiabatic expansion of steam from boiler pressure and initial quality conditions to exhaust pressure. (See Art. 185, Chap. XI) It is a simple matter, for any given condition, to compute the heat units that such a cycle renders available per pound of steam assumed to be operating, and if the quantity 2545 B.t.u. (which is the heat equivalent of 33,000 foot-pounds = 1 horse-power hour) is divided by the heat transformed into work per pound of steam, we evidently obtain the number of pounds of steam required by the turbine per horse-power hour if it were operating on the theoretical cycle. If, further, this theoretical water rate is then divided by the actual water rate of the turbine as determined from tests, we obtain an efficiency ratio which has been called *potential or Clausius efficiency*, and is analogous to the cylinder efficiency in reciprocating engines.

Let H = the heat units transformed into work per pound of steam operating on the theoretical Clausius cycle,

and

W = the actual water rate of the turbine in pounds per horse-power hour.

Then the

$$\text{Clausius efficiency} = \frac{2545}{H\,W}$$

H may be computed but is most easily determined by means of the Mollier diagram, Fig. 245.

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* These figures are slightly changed from the original, Marks' and Davis' Steam Tables being used.

Example. — In the above test figures, quoted from French, the condenser pressure is not given, but assume that the exhaust pressure was 1 pound absolute. For 170 pounds absolute pressure and 109 degrees of superheat, the diagram gives 1257 B.t.u. per pound of steam, as closely as it can be read. From the point in this diagram denoting this condition of steam, drop vertically downward (adiabatic or isentropic change) until the 1-pound absolute-pressure line is reached. The quality of the steam has dropped to 80.2 per cent, while the heat content is now 896 B.t.u. per pound. Thus the theoretical cycle renders available $H = 1257 - 896 = 361$ B.t.u. The theoretical water-rate is therefore $\frac{2545}{361} = 7.05$ pounds per horse-power hour. Since the actual water-rate based on the internal horse power was 11.67 pounds, the corresponding Clausius efficiency is $\frac{7.05}{11.67} = 60.4$ per cent.

375. Separation and Estimation of Steam-turbine Losses. — The losses occurring in steam turbines may be classified as follows:

1. Losses in nozzles and blades, owing to imperfect action of the steam in its passage. The greater part of this loss is in the blades, the nozzle efficiency being in most cases over 95 per cent.
2. Disk and blade friction, due to the rotation in the steam. Also windage losses in electric generator. (Together called rotation losses.)
3. Leakage of steam through clearance spaces without doing work.
4. Bearing and stuffing-box friction losses.
5. Loss due to radiation.

The sum total of these losses is measured by the difference between 100 per cent and the Clausius efficiency, based upon the brake horse power, *not* the internal horse power. Thus, in the figures above given, the equivalent mechanical efficiency $\frac{\text{B.H.P.}}{\text{Internal H.P.}}$ is assumed = 94 per cent. Hence, based upon the brake horse power, the Clausius efficiency in the above example would have been $.94 \times 60.4 = 56.7$ per cent. The sum of the five losses above enumerated is, then, $100 - 56.7 = 43.3$ per cent.

A first attempt at separating the individual losses may be made on the basis of Willan's line. In Fig. 520, let *AB* be the curve of total steam consumption per hour. Compute the data for the line *OC*, which is the line of total steam consumption for the turbine operating on the theoretical Clausius cycle. Since there are no losses, this line will pass through the origin *O* for no load. Through *O* draw also the line *OB'* parallel to *AB*. Prolong the line *BA* to *D*.

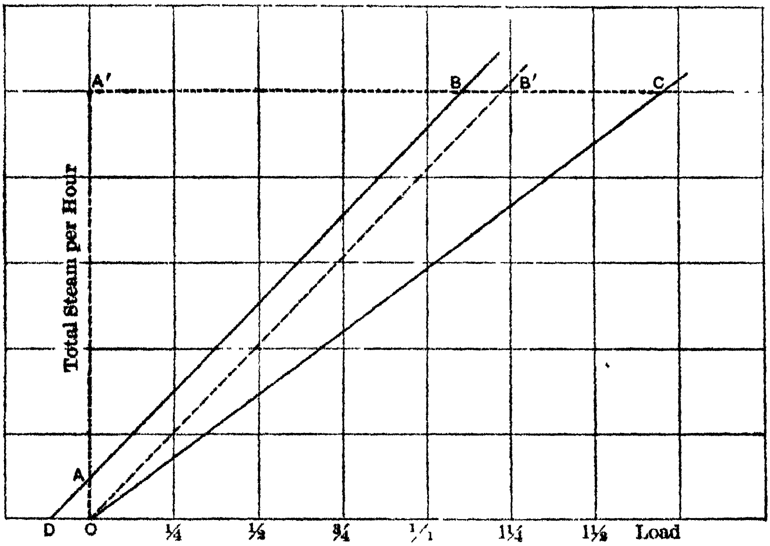


FIG. 520.—DETERMINATION OF TURBINE LOSSES.

The line *AB* shows that there is a certain quantity of steam, represented by *OA*, used per hour when the turbine is delivering no power and simply turning over against its own resistances. The power thus expended and lost is measured by the distance *OD*, and is represented by losses 2, 4, and 5, enumerated above. These losses are fairly constant and independent of the load. Hence they can be eliminated by drawing in the line *OB'*. The rest of the losses may be considered varying with the load, and are represented in the diagram by the widening distances between lines *OB'* and *OC* as the load increases.

To determine the percentage of the losses at any given load, say *B*, draw through *B* the line *A'C*. Distance *A'C* shows the power that the steam used should have produced had there been no losses. Distance *A'B* is the power actually produced. Hence, $\frac{A'B}{A'C}$ = the Clausius efficiency. The distance *BC* represents total power lost. Of this the part *BB'* is due to the constant losses and *B'C* to the variable losses.

The further subdivision of the two classes of losses is not a simple matter and the reader is referred to steam-turbine treatises for detailed information.

376. Method of Correcting the Results of Steam-Turbine Tests for Standard or Guarantee Conditions. — Contract guarantees generally state that a steam turbine will operate on not to exceed a certain steam consumption per K.W.- or B.H.P.-hour for a certain absolute steam pressure at the inlet, a certain quality of steam (usually superheat), and a certain vacuum. It occasionally happens that on an acceptance test one or the other of the conditions specified cannot be exactly met, and the question then arises as to what allowances shall be made mutually satisfactory to manufacturer and purchaser. The same question also comes up when it is desired to compare turbines of the same size but of different types with respect to steam economy, if the test figures available were not obtained under the same conditions of pressure, superheat, and vacuum.

The method of correcting is in both cases the same. In the former case the test figures are corrected to the guarantee conditions, while in the latter case some standard set of conditions (usually about the average of two sets of actual conditions) is assumed and all the results are computed to this standard. In either case it is of course necessary to obtain certain data on the change of economy with a change of any of the three conditions named. Such data can, in nearly all cases, be obtained from the manufacturers. In contract guarantees it is common for the manufacturers to give tables showing the allowances to be made in specific cases.

How the corrections are computed for full load is perhaps best shown by a concrete example. The figures are quoted from Moyer, "Steam Turbines," Chap. VI. Fig. 521 shows the three correction curves for full load furnished by the manufacturer for a 125-K.W. turbine.

This turbine, with 175 pounds absolute admission pressure, 27.5 inches vacuum, and 50 degrees of superheat, showed a steam consumption at full load of 24.5 pounds per kilowatt-hour. It is desired to recompute this to the basis of 165 pounds absolute pressure, 28 inches vacuum, and 0 degrees superheat.

Curve I, Fig. 521, between 27 and 28 inches vacuum, shows a gain of 1.0 pound in steam consumption, hence the correction should be a subtraction of .5 pound from the observed steam consumption, since the new basis shows an increase of ½ inch vacuum.

Curve II shows a gain of 2 pounds in the steam consumption for an increase in superheat from 0 to 100 degrees. The correction should therefore be an addition of 1 pound to the observed steam consumption, because the new basis shows a loss of 50 degrees of superheat.

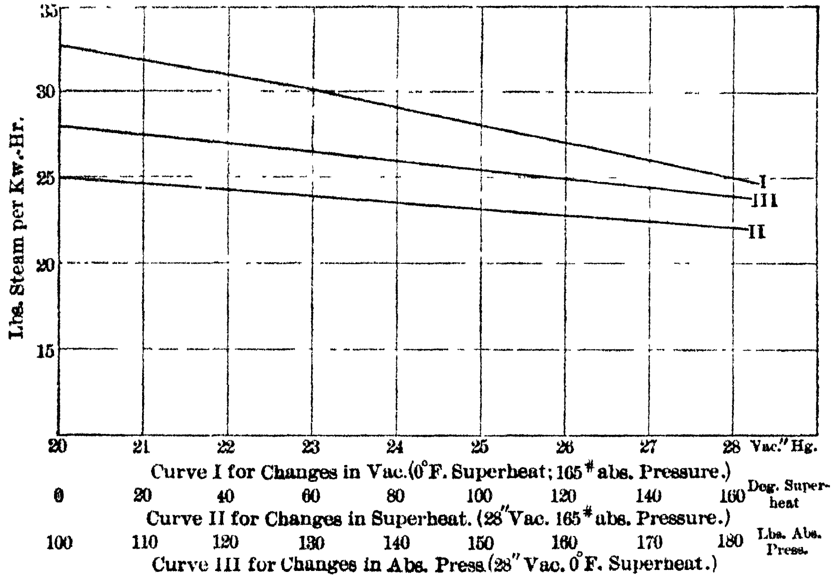


FIG. 521. — CURVES SHOWING VARIATION OF TURBINE ECONOMY.

Finally, Curve III shows a gain of 4 pounds in the steam consumption for an increase of 80 pounds in the admission pressure. Hence the correction should be an addition of .5 pound to the observed water-rate, since the new basis shows a loss of 10 pounds admission pressure

The corrected steam consumption therefore is 24.5 − .5 + 1.0 + .5 = 25.5 pounds per kilowatt-hour.

Correction for *fractional loads* (¼ load to 1½ load) may be made in a similar manner by a "ratio" or "percentage" method, on the assumption that the percentage correction found at full load applies also at any other load. This is very nearly true for any but the very low loads.

To explain the method, it will be found from Curve I that the consumption for 27.5 inches vacuum is 25.6 pounds, while for 28 inches vacuum it is 25 pounds per kilowatt-hour. Obviously the percentage correction, from the 27.5 inches to the 28 inches basis, is $\frac{25.6 - 25}{25.6} = -2.34$ per cent. This correction is negative (that is, a subtraction) because the new basis of computation shows a better vacuum and hence better economy, as already stated. Similar percentage factors may, from Curves II and III, also be computed for changes of superheat and of admission pressure, keeping in mind always to designate the factors by a minus sign if the change from the actual to the assumed basis is a gain of economy, and using the plus sign if the change is accompanied by an increase in the steam consumption. For this particular case the following table shows the final result:

	Test Conditions.		Assumed Conditions.		Correction Ratio.	Correction Percentage.
		Steam Cons. Lbs. per K.W.-hr.		Steam Cons. Lbs. per K W.-hr.		
Vacuum, inches.......	27.5	25.6	28.0	25.0	$\frac{25.6 \times 25.0}{25.6}$	−2.34
Superheat, deg. F....	50.0	24.0	0.0	25.0	$\frac{25.0 \times 24.0}{24.0}$	+4.17
Abs. press. adm..... .	175.0	24.3	165.0	24.8	$\frac{24.8 - 24.3}{24.3}$	+2.06
					Net correction...	3.89

Finally, the following table shows the results obtained for this turbine at fractional loads and the percentage correction applied to them:

	1/4	1/2	3/4	Full.	1 1/4
Steam consumption from tests, lbs per K.W.-hr.	31.2	26.9	25.2	24.5	23.6
Net correction, +3.89%........	1.2	1.1	1.0	1.0	0.9
Corrected steam consumption, lbs. per K.W.-hr.	32.4	28.0	26.2	25.5	24.5

CHAPTER XX.

THE TESTING OF INJECTORS.

377. General Theory of the Instrument.—An injector is an apparatus in which a jet of vapor from a nozzle strikes upon, and is condensed by, a mass of liquid, with the result that the combined mass of liquid and condensed vapor gains kinetic energy comparable with that which a similar liquid mass would gain by discharge through a nozzle under the same pressure as that driving the vapor jet. The underlying principle which makes the injector possible is that the velocity of discharge of a gas or vapor from a nozzle, under a certain difference of pressures through the nozzle, is considerably greater than the corresponding velocity for a liquid under the same pressure conditions. The greater velocity for the gas or vapor arises from a transformation of part of the heat energy content of the gas or vapor into mass-velocity energy, a transformation that either does not occur at all, or only in slight amounts, for liquids. This extra velocity energy of the vapor then becomes available to force both the condensed vapor and some additional condensing liquid from the lower to the higher pressure. The vapor jet and the condensing liquid may be different substances; in practice, however, the vapor is *steam* and the condensing liquid *water*.

The necessary parts of an injector are three: the steam nozzle, the combining tube, and the delivery tube. See Fig. 522.

The steam nozzle should so control the flow of the steam supplied, that in dropping the pressure from the supply pressure to the pressure of the combining space, the steam should gain the maximum possible velocity; that is, the nozzle expansion should be isentropic.

The pressure in the combining space, the lower of the two pressures between which the nozzle works, cannot be lower than, and for perfect action should be as low as, the vapor pressure determined by the temperature of the suction water entering the combining space. This definition of the theoretical pressure in the combining space is of the utmost importance in the determination both of the efficiency and of the limits of operation of the injector.

In the combining tube, the energy exchange of velocities between water and steam is by impact. Hence the flows of water and steam, as they come together, should be as nearly as possible in the same direction; any cross velocities result in loss of efficiency of impact. Low efficiency of impact means that the kinetic energy of the steam is spent in heating the water rather than in giving it kinetic energy. The further function of the combining tube is the condensation of the steam in the water. The steam, in the usual case of being initially nearly dry and saturated, decreases in quality during the isentropic expansion in the steam nozzle, coming from the nozzle some 10 or 15 per cent wet. The condensation is not *completely* finished in the combining tube. It cannot be complete until sufficient velocity head of flow is changed to pressure head that the pressure equals or exceeds the boiling pressure corresponding to the temperature of the fluid. The maximum velocity of flow, and hence the lowest pressure head, occurs at the end of the combining tube and beginning of the delivery tube; the pressure necessary for complete condensation of the steam is therefore not attained until the fluid mass is part way through the delivery tube.

The function of the delivery tube is to change velocity head to pressure head. The fluid enters the delivery tube with very low pressure and very high velocity (see the pressure and velocity diagrams in Fig. 522); it leaves the tube with pressure equal to the discharge pressure and almost negligible velocity. In the delivery tube, as in the steam nozzle, the first condition for highest mechanical efficiency (lowest friction loss) is that the change of section area along the length of the tube shall be such as to give uniform acceleration to the fluid handled. The second condition is that the tube shall be as short as possible with avoidance of eddy currents in the stream; this reduces skin friction to a minimum. As explained above, the completion of condensation of the steam occurs in the delivery tube.

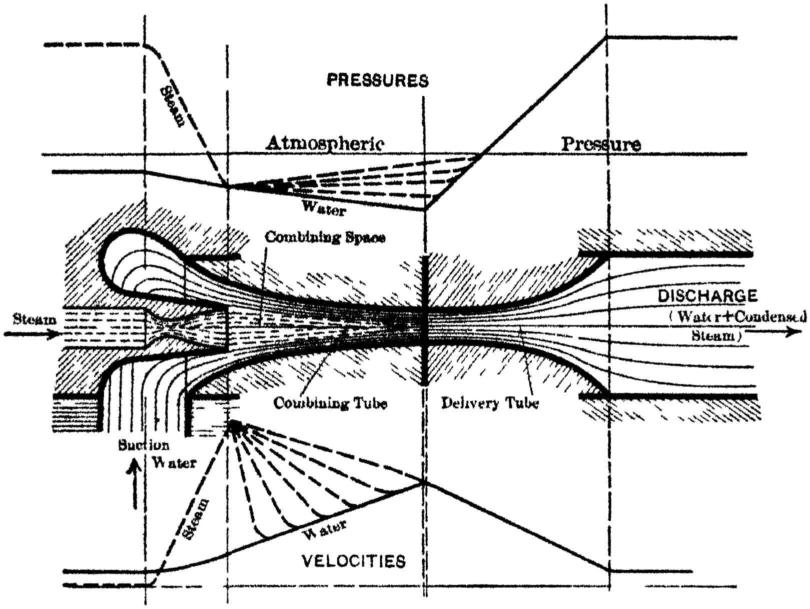


FIG 522 —DIAGRAM OF INJECTOR

378. Practical Construction and Operation. A distinction may be made between *injectors proper*, which are intended to discharge against a pressure equal to or greater than the steam pressure, and *ejectors*, discharging against a lower pressure than that of the steam. Injectors proper handle correspondingly smaller amounts of water than do ejectors.

Injectors now on the market may be classified as:

1*a*. Automatic or restarting, when the mere opening of the steam valve is all the handling necessary to bring the instrument into operation.

1*b*. Nonautomatic, when manipulation of steam, water, and overflow valves is necessary each time the injector is started.

2*a* and 2*b*. Single tube or double tube, depending upon whether the instrument has between the suction and discharge ends one single set of steam nozzle, combining tube, and delivery tube, or two such sets in series with each other. Each type may be automatic (2*a*) or nonautomatic (2*b*)

3*a* and 3*b*. Lifting or nonlifting, according as the instrument can, in starting, pick up its supply water through an appreciable suction head, or requires that the supply water be received at the instrument under some pressure. Each type may again be automatic (3*a*) or nonautomatic (3*b*).

4*a*. Live steam injectors draw their steam directly from the boiler.

4*b*. Exhaust steam injectors operate on exhaust steam from some steam user such as an engine or pump. Exhaust steam injectors usually require auxiliary live steam supply.

5*a* and 5*b*. Compensating or noncompensating, depending upon whether the relative positions or sizes of the various nozzles and tubes may be adjusted to best suit various operating conditions, or whether these parts are fixed in size and position. Compensation may or may not be automatic.

1*a*. Automatic starting is secured by using some form of divided combining tube. When steam is turned on it flows through the steam nozzle and the first part of the combining tube, then escapes into the overflow space, and from this space into the air through a large and easily moved check valve. Flowing in this way, the steam rapidly sucks the air from the suction water line, and draws water into the instrument. As soon as enough water enters to

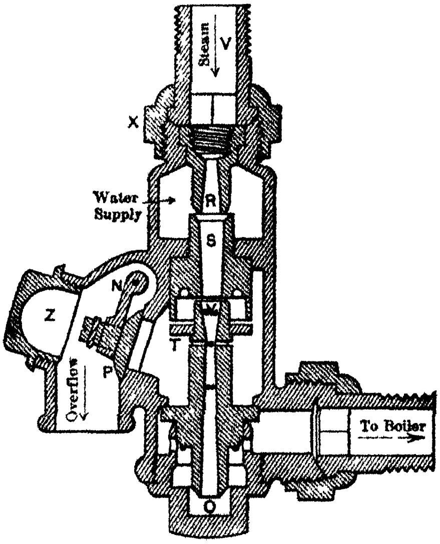


Fig. 523.—Single-Tube Automatic Lifting Injector (Penberthy).

thoroughly condense the steam, the combined mass flows through the delivery tube and passes the delivery check valve. The pressure in the overflow space then drops to less than atmospheric, and pressure relations along the combining and delivery tubes are used to cause an automatic closing together of the hitherto divided parts of the combining tube. An automatic injector will restart itself after any temporary interruption of action, provided the steam is left on.

1*b*. Nonautomatic injectors generally start by manipulation of the size and position of the steam nozzle. The overflow valve may or may not be automatic.

2. Double-tube injectors use one set of tubes for lifting and one for forcing. They have, in general, a wider operating range than single-tube injectors.

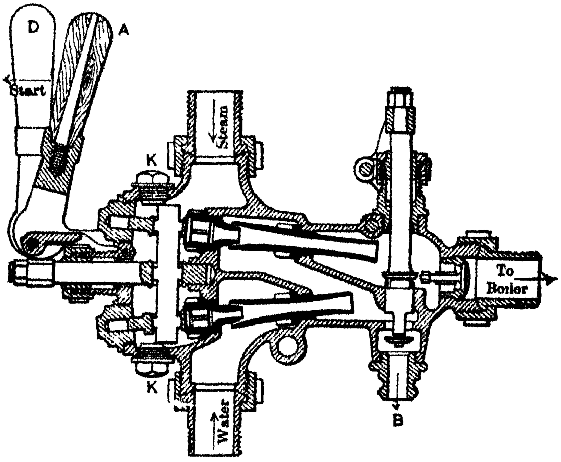


Fig. 524.—Double-Tube Nonautomatic Lifting Injector (Schütte-Koerting)

3. The sizes, forms, and relative positions of steam nozzle and combining tube determine whether an injector is lifting or nonlifting.

4. Exhaust steam injectors are usually double-tube injectors, using exhaust steam in the lifting set of tubes, and exhaust steam with or without auxiliary live steam in the forcing set of tubes. Whether or not live steam is necessary is determined by the pres-

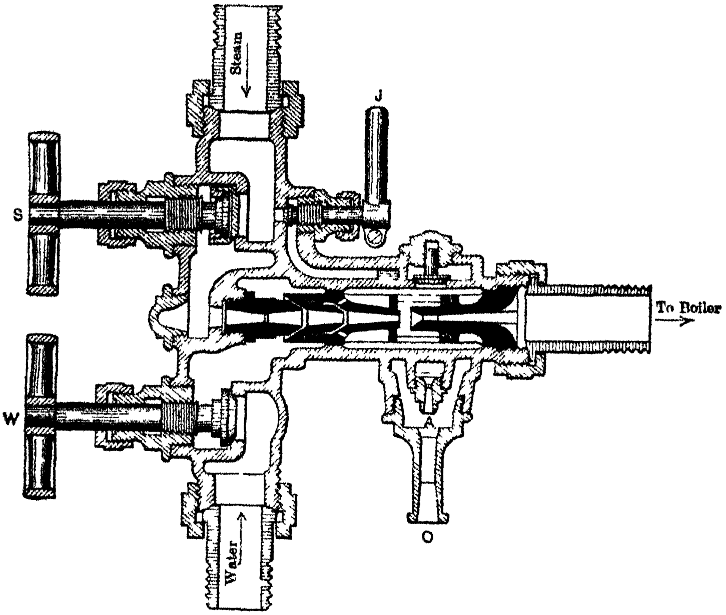
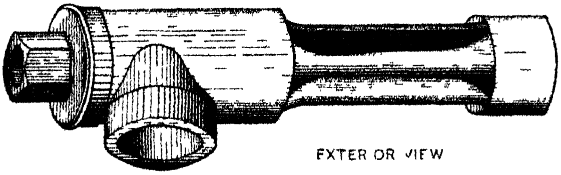
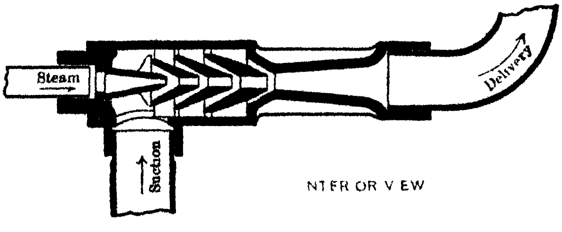


FIG. 525 — SINGLE TUBE NONAUTOMATIC LIFTING INJECTOR (MONITOR)



EXTERIOR VIEW



INTERIOR VIEW

FIG. 526. — AUTOMATIC LIFTING EJECTOR

sure of the exhaust steam and the pressure against which the discharge must go

5. Nonautomatic injectors are generally more or less compensating, through possible changes in relative positions of parts Double-tube injectors are partially compensating because of the differentiation of functions

Figs. 523, 524, 525 and 526 illustrate various forms of injectors. Fig. 525, by the omission of the ejector nozzle *A* and valve *J* and their connecting passages, becomes a nonlifting injector.

379. Efficiency of Injectors. — The injector may be considered in two ways: as a mechanical pump, and as a device for supplying hot feed water.

Viewing the injector merely as a pump, the energy input is measured by the weight of steam supplied in unit time multiplied by the heat energy in one pound of steam above 32° F., under the observed conditions. The output is measured by the weight of discharge water multiplied by the total head through which it is lifted. In this view of its operation the injector is analogous to a piston steam pump which condenses its exhaust steam in its suction water, and discharges the combined mass of suction water and condensed steam. No credit is here given the injector for the fact that the water is heated as well as moved.

When used as a boiler-feeding device, or as a pump where the heating of the water handled is an object, the injector must be credited with both the pump work done and the heating effected. The only loss is then the so-called " radiation loss " of heat from the injector and its piping, as all friction losses in the flow of steam or water reappear as heat in the discharge. As this " radiation " is quite negligible in comparison with the heat quantities in the steam, the heat efficiency of the injector is very nearly 100 per cent.

Let w = pounds of steam supplied per unit of time.

H = the " total heat " per pound of such steam as supplied $= xr + q$ or $\lambda + C_p D$.

v = velocity of the steam, ft. per sec., at the point where H is determined.

h_w = distance from injector to center of steam gauge. See Fig. 527.

J = the mechanical equivalent of heat = 777.5.

Then the total energy supplied to the injector in steam in unit time is

$$wH + \frac{wh_w}{J} + \frac{wv^2}{2gJ}\text{B.t.u.,} \qquad (1)$$

or
$$wHJ + wh_w + \frac{wv^2}{2g}\text{ ft.-lbs.} \qquad (2)$$

Let W = the pounds of suction water in unit time.

p_c = the absolute vapor pressure in lbs. per sq. in. corresponding to the suction water temperature.

p_d = the discharge pressure, lbs. per sq. in. abs.

h_d = the head in ft., down to the injector, from the center of the discharge gauge. See Fig. 527.

p_s = the suction pressure, lbs. per sq. in. abs.

h_s = the head in ft., from injector down to the center of the suction gauge. See Fig. 527.

V = the discharge velocity, ft. per sec., at point where p_d is measured.

δ = the density of the discharge water in lbs. per cu. ft.

Then useful work done in lifting suction water is

$$W\left(h_s + h_d + \frac{V^2}{2g} + \frac{p_d - p_s}{\delta} \cdot 144\right)\text{ft.-lbs. in unit time,} \qquad (3)$$

and useful work done in lifting condensed steam is

$$W\left(h_d + \frac{V^2}{2g} + \frac{p_d - p_c}{\delta} \cdot 144\right)\text{ft.-lbs. in unit time.} \qquad (4)$$

Note that in these equations p_d, p_s, and p_c are expressed in pounds per square inch *absolute*.

The *pump efficiency* of the injector is the sum of (3) and (4) divided by (2), or

$$\frac{W\left\{h_s + h_d + \frac{V^2}{2g} + \frac{144\,(p_d - p_s)}{\delta}\right\} + w\left\{h_d + \frac{V^2}{2g} + \frac{144\,(p_d - p_c)}{\delta}\right\}}{wHJ + wh_w + \frac{wv^2}{2g}}. \qquad (5)$$

Let q_s = the sensible heat of water at the suction temperature.

q_d = the sensible heat of water at the discharge temperature.

Then the heat accounted for already in the suction water is Wq_s; the heat accounted for in discharge is $(W + w)\,q_d$. The difference is $(W + w)\,q_d - Wq_s$. Note that this heat increase includes all the mechanical friction losses, such as friction head losses in suction and discharge, and impact losses in the injector. If the injector be credited with this heat gain as well as with the useful mechanical work done, we have as the *heat efficiency* of the injector

$$\frac{W\left\{h_s+h_d+\frac{V^2}{2g}+\frac{144\,(p_d-p_s)}{\delta}\right\}+w\left\{h_d+\frac{V^2}{2g}+\frac{144\,(p_d-p_c)}{\delta}\right\}+J\left\{(W+w)q_d-Wq_s\right\}}{wHJ+wh_w+\frac{wv^2}{2g}} \quad (6)$$

In the above equations, as circumstances change, various quantities may become negligibly small. The one quantity which will almost always be negligible is the term wh_w in the denominators of Eqs. (5) and (6).

Equation (6) should give practically 100 per cent. The magnitude of the "radiation" loss from an injector and its piping should be of the order of 0.1 to 0.5 of one per cent of the heat supplied in the steam.

380. **Testing of Injectors.**—One type of testing apparatus is shown in diagram in Fig. 527. It is based upon one designed and described by Schrauff in the *Zeitschrift des Vereines deutchser Ingenieure,* some modifications being made. The idea is to have continuous

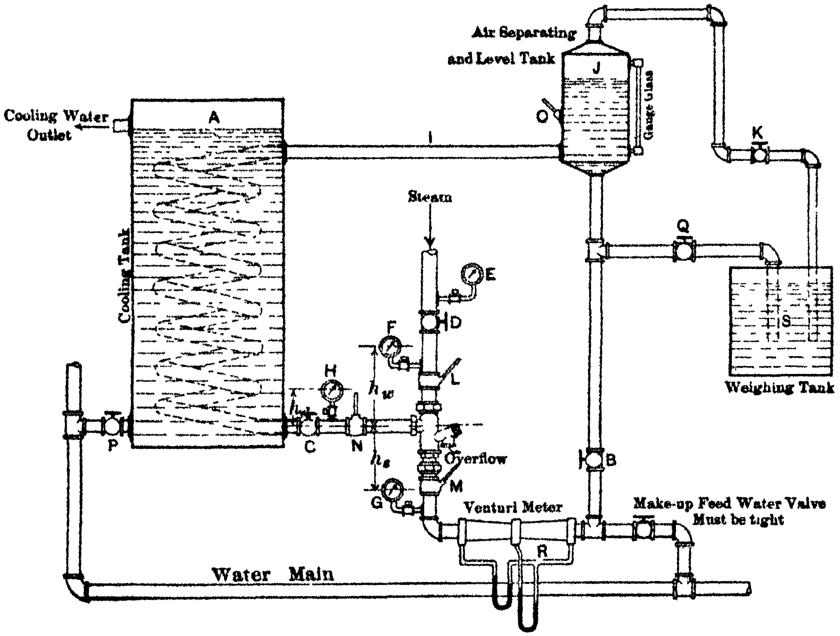


FIG. 527.—APPARATUS FOR TESTING INJECTORS.

flow in a closed circuit. In passing through the injector, the water experiences both a pressure and a temperature increase. The temperature is again brought back to the initial by means of the cooling tank *A*, while the pressure is reduced and controlled by a valve *B* in the suction pipe to the injector. The discharge pressure is controlled by valve *C*. The steam pressure on the injector is regulated by valve *D*. The quality of the steam below valve *D* must be determined. Four gauges, *E*, *F*, *G*, and *H*, of which *G* should be a compound gauge, serve to measure the main steam pressure, the steam pressure on the injector, the suction pressure, and the discharge pressure, respectively. The quantity of water flowing through pipe *I* consists, of course, of the sum of suction water plus condensed steam. This flows into a tank *J* which serves both as an air-separating tank and as a " steam-level " tank. The air which separates goes to the top of the tank and may be allowed to escape through the air valve *K*.

Temperatures are read at *L* (steam), *M* (suction water), *N* (discharge water), and *O* (in level tank).

To make a run, start the injector and adjust valves *D*, *B*, *C*, cooling water control valve *P*, and condensed steam control valve *Q*, so that the desired pressure and temperature conditions are established, and so that the water level in tank *J* remains at one point as noted on the gauge glass. When constant conditions are established, the run may be started and continued as long as desired. The actual quantity of suction water handled may be determined by any good meter. In the diagram, a Venturi meter *R* is indicated. The condensed steam is caught and weighed in tank *S*. To make certain that there is no loss at this point from vaporization, it is well to have both the condensed steam and the air pipe dip into cold water.

Rough testing of an injector may be done with a pair of similar tanks on scales, drawing suction water from one tank and discharging into the other. The suction and discharge piping running down into the tanks must then be of the same size, and end fittings the same, to avoid displacement errors in weights. The suction tank readings give the rate W; the discharge gives $(W + w)$. The values of w so obtained are subject to large errors. It is perhaps better to compute w from the observed value of W by means of Eq. (6), on the assumption that heat efficiency = 100 per cent.

The independent variables entering into the injector operation are:

(1) Steam pressure and heat content of steam.
(2) Suction pressure.
(3) Suction temperature.
(4) Discharge pressure.

When an injector is used as a boiler feeder (1) and (4) are equal; more precisely, (4) just exceeds (1).

An injector test should include the following runs:

1. With suction pressure and temperature constant at average values, take a series of steam pressures and for each steam pressure vary the discharge pressure from the lowest to the highest under which the injector will operate. Between the lowest and highest discharge pressure for each steam pressure, choose three or four other discharge pressures in a series in order to obtain five or six points for a curve.

2. With steam and discharge pressures adjusted at any desired points (usually taken as the pressures of regular ordinary operation), and for an average suction lift or pressure, make runs with a series of temperatures of suction water, continuing to increase the temperature until the injector "breaks," that is, refuses to operate, the discharge either going out of the overflow or the steam blowing down the suction pipe. Make sufficient runs to determine a curve.

3. With the same steam and discharge pressures as under 2, and with an average suction temperature, make a series of runs with varying suction lifts or pressures, increasing the lift until the injector breaks.

Of the following forms, the first may be used for recording the observations, the second shows the principal data that should be reported, together with the items to be obtained by computation.

The best method to show the limits of operation of the injector, efficiency, etc., is to plot certain curves. The most useful of these are the following:

(*a*) For the suction pressure and temperature under (1) above, plot as abscissæ steam pressures, and as ordinates

1. Values of w.
2. Values of W, maximum and minimum, or
3. Values of $\frac{w}{W}$, maximum and minimum, also
4. Values of maximum discharge pressure, and
5. Values of minimum discharge pressure.

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Test of .... *Injector* *Made* .... 19. .
Dia. Suction Pipe ...... *Dia. Discharge Pipe* .. *By* ....
Dia. Steam Orifice .... *Dia. Water Orifice* .. . .

No.	Time.	Barometer.	Gauges, Lbs				Temperatures, Degrees F.				Lbs of Suction Water.	Lbs of Steam
			Boiler.	Injector.	Delivery	Suction.	Supply	Delivery.	Calorim.	Room		

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DATA AND RESULTS OF INJECTOR TEST.

Test of .... *Injector* *Date* .... 191....

Vertical Distance from Injector to Center of Steam gauge = h_w *ft.*........ *Made by*...............

Vertical Distance from Injector to Center of Discharge gauge = h_d *ft.*........ *Barometer Reading* .....

Vertical Distance from Injector to Center of Suction Gauge = h_s *ft.*........

Run No.	1	2	3	4	5	6
Duration of test, minutes						
Boiler pressure, by gauge						
Steam pressure on injector, by gauge						
Steam pressure on injector, absolute						
Suction pressure, by gauge						
Suction pressure, absolute, $=p_s$						
Discharge pressure, by gauge						
Discharge pressure, absolute, $=p_d$						
Pressure corresponding to suction water temp., absolute, $=p_c$						
Total discharge head in feet						
Total suction head in feet						
Steam supplied per hour $=w$						
Suction water per hour $=W$						
Ratio $W \div w$						
Velocity of steam at point where pressure is measured, ft., per sec., $=v$						
Velocity of discharge at point where pressure is measured, ft., per sec., $=V$.						
Density of discharge water, lbs. per cu. ft. $=\delta$.						
Quality of steam, moisture, or superheat						
Sensible heat of water at suction temp., B.t.u., $=q_s$						
Sensible heat of water at discharge temp., B.t.u., $=q_d$						
Total heat per pound of steam supplied, B.t.u., H						
Total energy supplied to injector per hour, B.t.u.						
Total energy supplied to injector per hour, ft.-lbs.						
Useful work done in lifting suction water, B.t.u.						
Useful work done in lifting suction water, ft.-lbs.						
Useful work done in lifting cond. steam, B.t.u.						
Useful work done in lifting cond. steam, ft.-lbs.						
Pump efficiency of injector, per cent						
Heat efficiency of injector, per cent						

(*b*) With steam and suction pressures constant at the probable working values, plot as abscissæ suction temperatures, ordinates same as under (*a*).

(*c*) With steam pressure and suction temperature constant at the probable working values, plot as abscissæ suction pressures, ordinates same as under (*a*).

If all of the curves listed above should be determined and plotted, they would show the operation of the instrument in so complete a fashion as to answer any of the questions which would be likely to arise in any commercial practice.

An injector is peculiarly sensitive to variation of suction temperature. This sensitiveness will be explained by considering the heat energy available in isentropic drop from the initial condition of the steam to the vapor pressure corresponding to the temperature of the suction water. It will be found that the available heat energy falls very rapidly as the suction temperature rises.

CHAPTER XXI.

GAS ENGINES AND GAS PRODUCERS.

381. Types of Gas Engines. — All gas engines belong to the class of internal combustion engines. (See introductory paragraph, Chap. XXII.) Without reference to any types that may have been tried and abandoned during the past, we, at the present day, distinguish only two fundamentally different types of internal combustion engines. The distinction is based upon how the combustion proceeds, whether at constant volume or at constant pressure, and we therefore have, as a primary classification, *constant volume* and *constant pressure* engines.

In engines operating upon the constant volume cycle, the combustible charge is compressed and then exploded at the inner dead center position of the piston. This results in practically constant volume combustion. The cycle is completed by the expansion stroke following, the burned gases being discharged after the opening of the exhaust valve, largely at constant volume. This is the theoretical cycle, which, from the inventor who first applied it successfully in commerce, is also called the *Otto cycle*. For the

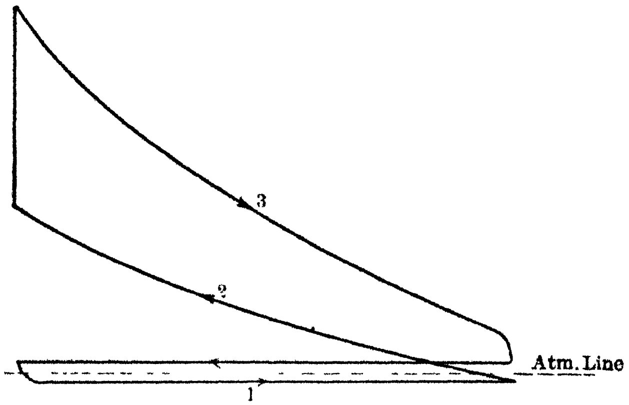

Fig. 528.—Typical 4-cycle Diagram.

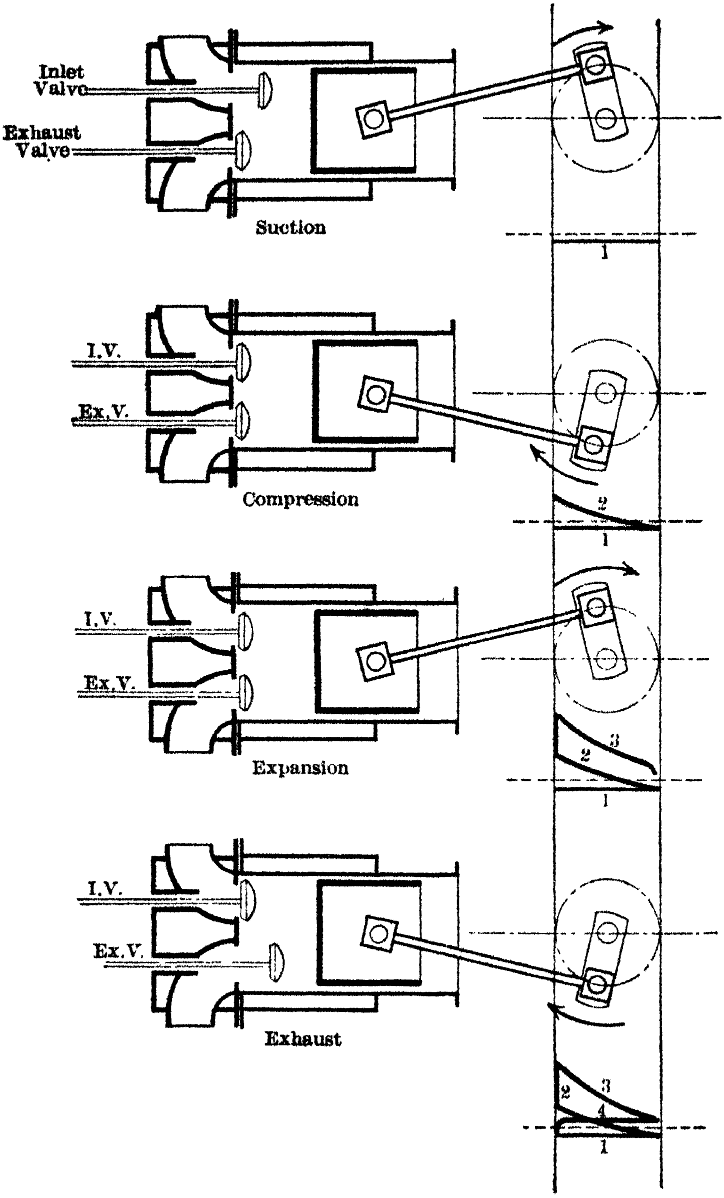

FIG. 529.—OPERATION OF 4-CYCLE ENGINE.

efficiency computations on the basis of this theoretical cycle see p. 349.

In practice, engines operating upon the constant volume or Otto cycle are divided into two classes, depending upon the method of charging and discharging the cylinder. We distinguish accordingly 4-cycle and 2-cycle engines. The terms four-stroke cycle and two-stroke cycle engines would be more strictly correct, but are too cumbersome, hence the shorter designation.

In 4-cycle engines, the charging and discharging actions are carried on by the engine cylinder itself, the latter acting as a pump for two strokes. Fig. 528 shows a conventional 4-cycle indicator diagram, the lower part of the diagram, the so-called lower loop, being somewhat exaggerated for the sake of clearness. The strokes are marked in order. Stroke 1 is the *suction* stroke, the piston on its outstroke drawing in the new charge behind it. Stroke 2 is the *compression* stroke. At the end of this instroke the charge is exploded, the pressure increasing at constant volume. Stroke 3 is the *expansion* stroke. Near the end of this stroke the exhaust valve opens and on the next instroke, the *exhaust* stroke, the piston drives the burned gases out ahead of itself, after which stroke 1 is repeated. The corresponding valve movement may be studied by aid of Fig. 529.

In a 2-cycle engine, the suction and exhaust strokes of the 4-cycle engine are eliminated, the cylinder being charged with fresh mixture and cleared of burned gases by agencies other than the action of the power piston of the engine. The power cycle is complete in two strokes and there is no lower loop to the diagram, the work of charging and discharging represented by the loop area being done in an outside pump of one type or another. Fig. 530 shows the typical 2-cycle diagram. The mixture, compressed by the instroke

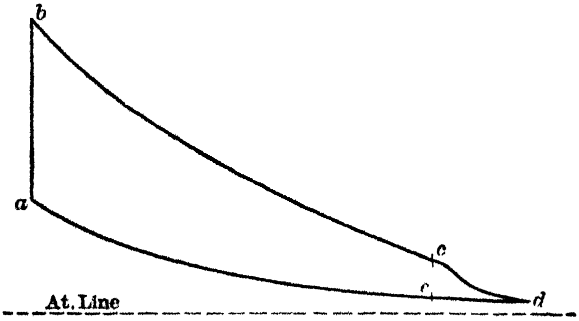

Fig. 530.—Typical 2-cycle Diagram.

of the piston, is ignited at point *a*. Expansion takes place from *b* to *c*. At *c* the exhaust ports open and the pressure drops rapidly to nearly atmosphere. Soon after point *c* is passed, the inlet valve opens and fresh mixture is forced in by the outside pump. At *d* the piston starts to return, and at *e* the exhaust port and inlet valve close; compression then begins. The charging and discharging actions evidently take place while the power piston is moving from *c* around *d* to *e*. The burned gases escape partly by their own expansion while the rest is forced out by the fresh mixture coming in. It is evident that the exhaust ports must close, whatever design of inlet and exhaust ports is used, before the combustible mixture reaches the exhaust ports. Otherwise there will be a serious loss of fuel.

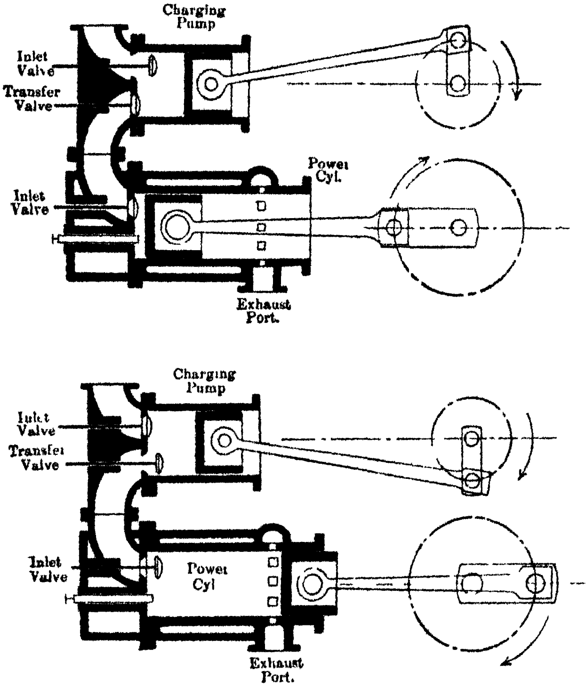

FIG. 531.—OPERATION OF 2-CYCLE ENGINE.

A conventional sketch of a type example of such an engine is shown in Fig. 531. It consists of a power cylinder with separate charging pump, whose crank leads the main crank by a certain amount (in this case about 90 degrees).

In the upper sketch the mixture is just being exploded and the piston starts on the outstroke (point *a*, Fig. 530). In the meantime the charging pump is drawing in the next charge. In the lower sketch the power piston has opened the exhaust ports (point *c* in Fig. 530), the hot gases have largely escaped, and the charging pump piston, being now on its instroke, is forcing a new charge into the power cylinder under some slight compression. When the power piston, on its return, closes the exhaust ports and the inlet valve to the cylinder also closes, compression begins (point *e* in Fig. 530).

In practice, nearly all medium or large sized 2-cycle engines are built with separate pumps as described. In some cases, to cheapen the construction, or to make it more compact, the front end of the power cylinder is enclosed, thus converting that end into a charging pump. A large variety of small, vertical, 2-cycle engines are built in which the entire crank mechanism together with the lower end of the cylinder are enclosed, and this space is used as a charging pump. Most small 2-cycle marine engines are constructed in this way. In both of the latter modifications the power piston also acts as the pump piston, but not in the power end of the cylinder.

The only commercial example of the constant-volume engine is the Diesel. Its operation is very similar to that described above for the constant volume type. The difference is that only air is compressed and that to the maximum pressure existing in the cycle. Then at or near the inner dead center, the liquid fuel, which is always used, is injected into this body of air, highly heated by compression. The result is combustion at practically constant pressure until the fuel valve cuts off. Then follow expansion and exhaust in the regular manner.

The student is referred to Chap. XVI, for type examples of actual indicator diagrams from gas and oil engines.

Gas engines may be made *single or double acting*. In a double-acting gas engine, each cylinder end is used for power development, the engine being built with piston rod, crosshead, and connecting rod, as in a steam engine. In this design, a 2-cycle engine will receive an impulse every stroke, just like a steam engine.

As far as *cylinder arrangement* is concerned, we distinguish the following:

Tandem. — All the cylinders are in line acting on the same crank. Not more than two cylinders are usually employed, and these may be either single or double acting.

Double, two-cylinder, three-cylinder engines, etc., according to the number. The cylinders are placed side by side, usually acting on as many cranks as there are cylinders. May be single or double acting.

Double-tandem. — Usually four cylinders, two each in tandem, acting on two cranks. May be single or double acting.

Opposed Arrangement. — Two or four cylinders opposed to each other, in the latter case in pairs. Any two opposed cylinders may act on the same crank, or each cylinder may have its own crank. Usually built single acting.

382. Gas Engine Fuels. — The combustible charge in a gas engine cylinder at the moment of explosion always consists of a mixture of a gas or of an oil vapor with certain quantities of air. The explosibility of these mixtures, as well as their characteristics in general, depend upon the proportion of fuel to air present. For a discussion of this matter, the student is referred to books on gas engines. It is desired to point out here merely the fact that the combustible part of a charge is always a gas or a vapor, irrespective of the original state of the fuel.

All three classes of commercial fuels, that is, the solid, the liquid, and the gas fuels, are used, but only the gas fuels can be used directly in an engine. The liquid fuels must first be vaporized or atomized in *carburetors*, *vaporizers*, or *spray nozzles*, while the solid fuels must be converted into *producer gas* in special apparatus called *producers* or *generators*.

The *gas fuels* most used in gas engine practice are: illuminating gas, natural gas, blast furnace gas, and producer gas. Any of these gases are fixed or permanent gases, and it is merely necessary to furnish the engine with suitable mixing valves to maintain the proper proportion between gas and air.

The *liquid fuels* are generally divided into two classes: the light or volatile liquids, and the heavy liquids. To the former belongs gasoline, while kerosene, the so-called distillates, and crude oils belong to the latter class. Alcohol holds a somewhat intermediate position. The distinction between the two classes is not very sharply defined. Generally it is stated that the light liquids can be successfully converted into fuel gas or vapor without the agency of heat, while the class of heavy liquid fuels usually requires heat. The use or nonuse of heat is apparently also the distinguishing mark between carburetors and vaporizers. Thus the apparatus used for the formation of the fuel gas in the case of gasoline is called a carburetor, the air merely passing through and taking up a sufficient quantity of the volatile liquid. In the case of alcohol, kerosene, or crude oil, this simple scheme is not applicable, these liquids not being sufficiently volatile at ordinary temperatures to form a combustible mixture with the air passing through the apparatus. It is necessary to apply heat to hasten volatilization (vaporization) and the apparatus used is then known as a vaporizer.

It is true that in some engines any of the liquids above mentioned are atomized or sprayed directly into the engine cylinder without first going through the process of volatilizing or vaporizing. It will generally be found upon examination, however, that (except perhaps in the case of the constant-pressure engines, like the Diesel) the fuel is sprayed into the cylinder at such a part of the cycle that sufficient time is available for the finely-divided oil " fog " floating in the charge of air to vaporize largely before ignition takes place.

The *solid fuels*, such as wood, peat, lignite, coke, and the various classes of coal, are gasified in producers, as has been already mentioned. The manufacture of producer gas has grown to be of great economic importance, so that the principles underlying the process merit some detailed discussion.

383. **Producer Gas.** — Producer gas is a composite gas consisting mainly of CO, H_2, CO_2, N_2, and certain other gases resulting from the distillation of the green fuel used. The producer-gas process is essentially a combination of the air- and water-gas processes. In the former, a fuel column is supplied with air alone under certain conditions of control, and the resulting gas consists of CO, CO_2, and N_2. In the water-gas process, water vapor (steam) alone is forced through a fuel column which has been previously made incandes-

cent by blowing air through it. The result is a gas consisting of CO, CO_2, and H_2. The reactions occurring during the time that water gas is being made absorb heat, and the process is combined with a continual cooling of the generator contents. The process of making water gas is consequently intermittent, a blowing period (with air, making air gas), alternating with a period of water gas make (blowing with steam). Combining these two periods, that is, blowing with a mixture of air and water vapor, makes the process continuous, and constitutes our modern producer-gas process.

The chemical reactions involved in the making of producer gas and the thermal relations existing are outlined by Fischer,* as follows. The quantities of heat given in each equation assume that *one pound* of carbon enters the reaction instead of stating the heat on the molecular weight basis, as is generally done in scientific works. The algebraic sign preceding the quantity of heat states whether heat is developed (plus sign) or absorbed (minus sign), that is, whether the reaction is exothermic or endothermic.

The first reaction occurring in the producer is probably the formation of CO_2 from the C in the fuel and the free oxygen of the air, according to the equation

$$C_2 + 2\,O_2 = 2\,CO_2 + 14{,}540\text{ B.t.u.} \qquad (1)$$

The development of heat indicated in (1) serves to render the fuel incandescent, so that with water vapor we next obtain the reactions

$$C_2 + 4\,H_2O = 2\,CO_2 + 4\,H_2 - 2848\text{ B.t.u.,} \qquad (2)$$

and

$$C_2 + 2\,CO_2 = 4\,CO - 5780\text{ B.t.u.} \qquad (3)$$

Both of these reactions are endothermic and require a temperature of about 1500° F. Equations (1) and (3) result in the indirect formation of CO. Combining these equations by addition to obtain a direct expression for this end result, we may write

$$C_2 + O_2 = 2\,CO + 4380\text{ B.t.u.} \qquad (4)$$

Similarly, a combination of equations (2) and (3) gives

$$C_2 + 2\,H_2O = 2\,H_2 + 2\,CO - 4316\text{ B.t.u.} \qquad (5)$$

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* Fischer, Kraftgas, Seine Herstellung & Beurteilung.

If there were no *losses* of heat, the *ideal producer-gas process* would then be a combination of equation (4) (ideal air-gas process) and equation (5) (ideal water-gas process), which may be written

$$2\,C_2 + O_2 + 2\,H_2O = 4\,CO + 2\,H_2. \qquad (I)$$

Since the oxygen in (4) is obtained from air, a certain amount of nitrogen (N_2) will also appear in both sides of equation (I). The composition, volume, and weight of the resulting gas per pound of C consumed is shown in the following table. To obtain volumes, the weights of the various gases per cubic foot at 14.7 pounds and 32° F. have been used.

	Weights Involved in Eq. I		Vol. of Producer Gas Cu. Ft. per Lb. of C	Composition of Gas	
	Left Side	Right Side		Per Cent by Weight	Per Cent by Volume
C_2	1.00				
O_2	.66				
N_2	2.23	2.230	28.4	48.0	38.9
H_2O	.75				
CO		2.330	29.8	50.2	41.0
H_2		.083	14.8	1.8	20.1
		4.643	73.0	100.0	100.0

In practice, however, a large part of the sensible heat developed by equation (4) is lost instead of being used to make up the deficiency shown in equation (5). Hence, if any given temperature level is to be maintained in the producer, the reaction represented by equation (4) must occur several times as rapidly in the same time as the one represented by equation (5). Assuming, for instance, that only 1500 B.t.u. of the 4380 in equation (4) are actually utilized, it is evident that (4) must occur about three times as often as (5) in the same time. The producer-gas process can then be represented by the equations

$$C_2 + 2\,H_2O = 2\,H_2 + 2\,CO - 4316\text{ B.t.u.}$$

$$3\,C_2 + 3\,O_2 = 6\,CO + 13{,}140\text{ B.t.u.}$$

or

$$4\,C_2 + 3\,O_2 + 2\,H_2O = 8\,CO + 2\,H_2 + 8824\text{ B.t.u.} \qquad (II)$$

If it is next assumed that the decomposition of the water vapor takes place solely according to equation (2), then, again assuming that only 1500 B.t.u. of the heat developed in equation (4) is utilized, the latter reaction would have to occur about twice as often in the same time as equation (2) to make up for the deficit of 2848 B.t.u. This condition can be represented by the equations

$$C_2 + 4\,H_2O = 2\,CO_2 + 4\,H_2 - 2848 \text{ B.t.u.}$$
$$2\,C_2 + 2\,O_2 = 4\,CO + 8760 \text{ B.t.u.}$$
$$\text{or } 3\,C_2 + 2\,O_2 + 4\,H_2O = 2\,CO_2 + 4\,CO + 4\,H_2 + 5912 \text{ B.t.u.} \quad \text{(III)}$$

In practice, gasification according to equations (II) and (III) is likely to go on simultaneously, so that in any given case the actual gas made will show a composition intermediate between those given for these two equations in the following table. The higher the temperatures, the more nearly will equation (II) be realized.

	Weights Involved in				Vol. of Prod Gas, Cu. Ft. per Lb. of C.		Composition of Gas			
	Eq. II.		Eq III.				Weight, Per Cent		Volume, Per Cent.	
	Left Side	Right Side	Left Side.	Right Side	Eq II	Eq III.	Eq II	Eq. III.	Eq. II	Eq III.
C_2	1		1.00							
O_2	1		89							
N_2	3.35	3.35	2.97	2.97	42.8	37.9	58.6	50.6	53.4	43.3
H_2O	.375		1.00							
CO		2.33		1.56	29.8	20.0	40.7	26.6	37.2	22.8
CO_2				1.23		10.0		21.0		11.4
H_2		042		111	7.5	10.7	.7	1.8	9.4	22.5
		5.722		5.871	80.1	87.6	100.0	100.0	100.0	100.0

384. Types of Gas Producers. — The simplest type of a gas producer is shown in Fig. 532. The fuel bed is divided into different zones showing where the reactions above outlined occur. The composition of the gas is, of course, changed from that computed by whatever gases or vapors are added to it in passing through the distillation zone.

It is obviously immaterial, as far as the gasification process is concerned, whether the air-steam mixture is forced through the fuel column by maintaining a slight pressure above atmosphere in the ash pit, or whether the mixture is drawn through by producing, by some means or other, a vacuum at the outlet of the producer, which vacuum, of course, causes a pressure difference between gas outlet and ash pit and hence produces flow. These two methods of producing gas flow, however, distinguish the two main types of producer-gas plant, the former being called a *pressure-gas*, the latter a *suction-gas* installation.

A pressure-gas plant requires a closed ash pit, and the gas is forced through an economizer, or preheater, and through a washing apparatus into a gas holder. The pressure is produced in various ways, generally by means of a steam blower. The essential parts of such a plant are shown in Fig. 533.* *B* is the generator with the filling hopper *C*. The air under some pressure, furnished by the blower *V*, reaches the ash pit by first passing through vaporizer (or economizer) *E*. The proper quantity of water is supplied to this vaporizer through the funnel *F*. The heat of the producer gas made, entering *E* through *n*, vaporizes the water and the vapor is picked up by the air passing on its way to the ash pit. The partly-cooled gas reaches the scrubber *G* through *W*, passing upward

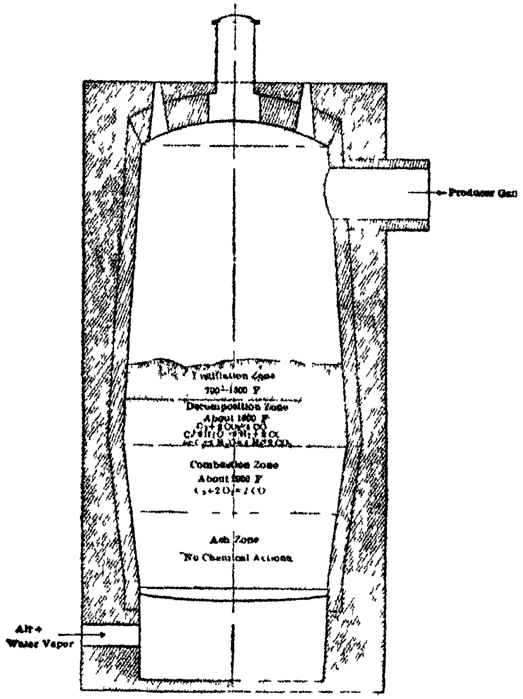

FIG. 532.—SIMPLE TYPE OF GAS PRODUCER.

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* This and the next three figures are due to Fischer, Kraftgas.

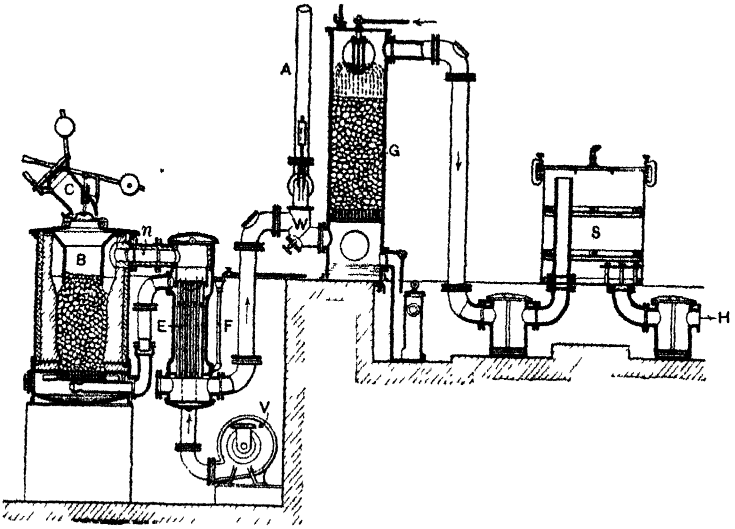

FIG. 533.—PRESSURE-GAS PLANT.

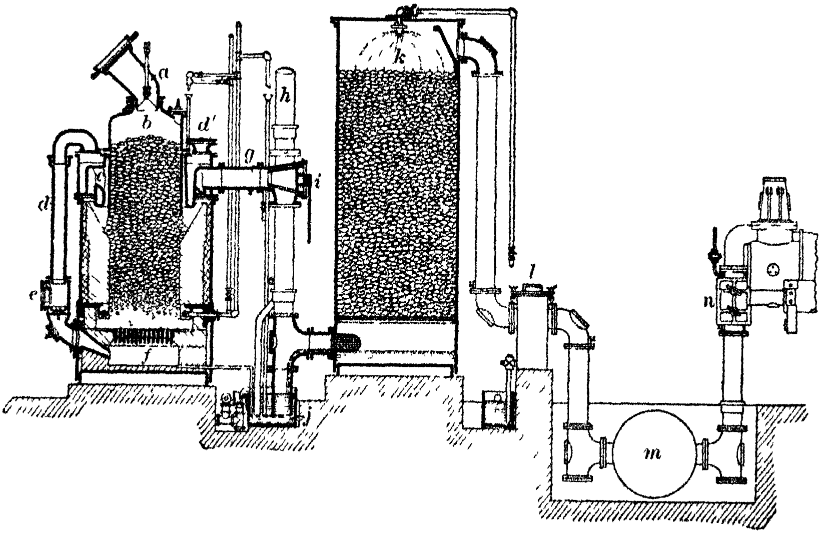

FIG. 534.—SUCTION-GAS PLANT.

through a coke column. The washing is done in *G* by water sprays passing downward against the ascending gas column. *A* is a purge pipe through which the poor gas made while the producer is being put into service is allowed to escape. The gas receives a final cleaning and drying in the purifier *S*, the trays of which carry either sawdust or excelsior, before being sent to the gas holder through *H*.

The main parts of a suction-gas plant are shown in Fig. 534. Here *b* is the producer with its filling hopper *a*. The vaporizer *c* is a hollow casting surrounding the top of the producer. As the piston of the engine cylinder makes an outward (suction) stroke, it creates a vacuum of several inches in the small receiver *m*. This causes a flow of gas toward the engine through seal-box *l*, scrubber *k*, and pipe *g*, and the difference of pressure thus produced between *g* and ash pit *f* causes a rush of air-steam mixture through the fuel column. The air enters the vaporizer through the regulating valve *d'*, saturates itself in passing through *c*, and finds its way through *d* into *f*; *e* is an auxiliary air-supply valve to regulate the proportion of air to water vapor; *h* is the purge pipe and *i* a double-throw valve; *n* is a " wire brush " cleaning box which serves to remove the last traces of tar that the gas may carry.

In gas-engine practice, the suction-gas plant has largely supplanted the pressure-gas installation, the main reasons being the smaller cost of installation, on account of the absence of the gas holder, and the fact that the regulation of the gas made in the suction-gas plant is automatic, the quantity made varying directly with the demands of the engine.

There are so-called "combination" plants. If, for instance, in Fig. 533 the blower *V* were removed and an exhauster had been installed at *H*, between the purifier and the gas holder, the installation up to the exhauster would evidently be of the suction type, while the gas beyond the exhauster would be under some pressure.

Producers are also classified according to the kind of fuel burned, and among the gases made we have anthracite-producer gas, bituminous-producer gas, peat gas, wood gas, oil gas, etc. Of course, for any given fuel, the plant may be either a pressure, a suction, or a combination gas plant, but in some cases the producers themselves show radical differences in design, depending upon the fuel used. These differences in design are necessitated by the fact that some of the fuels produce tar-forming gases to such an extent that the ordinary scrubbing apparatus is not capable of properly cleaning the gas for engine service. The types of producers above described will do very well for anthracite coal, a fuel which carries very little tar-forming gas (mostly hydrocarbons); and the gases of distillation escaping from the green fuel and which add themselves to the rest of the producer gas, therefore cause little trouble. The same is not true, however, of bituminous coal, and in that case special precautions must be taken to get a satisfactory gas. The way usually chosen is to collect the gases of distillation from the green fuel and to " fix " these gases, that is, to render them permanent in some manner. This has led to the design of " down-draft " and " double-zone " and " ring " (series) producers.

The principle involved is perhaps best understood by studying the series producer, Fig. 535. Two producers, G_1 and G_2, are connected as shown. In G_1 the fuel used is bituminous coal, in G_2 it is anthracite or coke. The gas in the connecting pipe y carries the tarry gases resulting from the distillation in G_1, and the entire body of gas is made to pass upward through the column of incandescent coke in G_2. An auxiliary air supply is furnished through

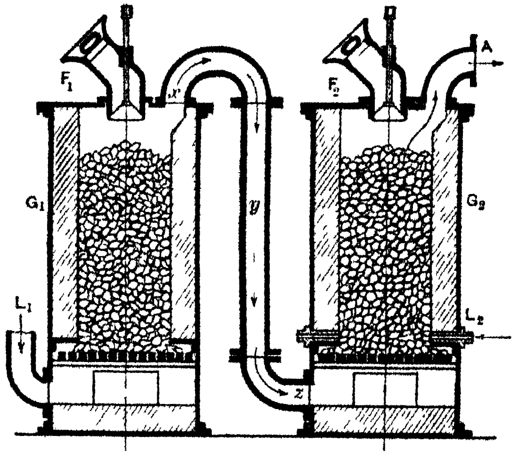

FIG. 535.—SERIES PRODUCERS

L_2. The tarry gases are either burned to H_2O and CO_2 near the base of the column, or certain reactions take place, the final products being H_2 and CO. These are, of course, permanent gases and the trouble of tar deposit is avoided.

In the down-draft producer the operation is very similar to that above described except that only one producer is used. The air-steam mixture is sent in at the top, under the filling hopper and moves downward through the fuel bed, carrying with it the gases of distillation from the green fuel in the upper layers. These gases are fixed, as before, in passing through the incandescent layers of fuel on their way to the bottom.

The double-zone principle is illustrated in Fig. 536, which represents a suction-gas producer. The green fuel is charged at the top and the fire near the top is maintained by an air supply through the cover. The vacuum produced at *c* draws the gases of distillation downward through the upper fire zone, fixing them. At the same time air enters at *a*, saturates itself in passing through the vaporizer, and finds its way through *b* to the ash pit, maintaining a second fire zone over the grate. The gas made passes out at *c*. The lower fuel supply is maintained by coked fuel which passes by the opening *c* unconsumed in the upper zone.

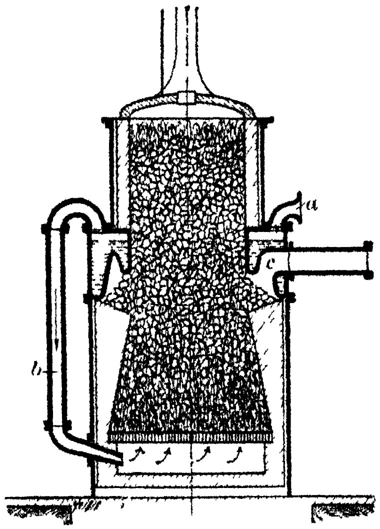

FIG. 536. — DOUBLE-ZONE PRODUCER.

There are many modifications of the design described, for which the reader must be referred to special books on the subject.

385. Data Relating to Gases, Fuels, etc., Generally Used in Efficiency Computations on Gas Producers and Gas Engines.

(*a*) *Atomic Weight, Density, Specific Weight and Volume. Standard Conditions.* — The *density* Δ of a gas is generally referred to air as a standard, and is defined as the weight of a certain volume of a given gas divided by the weight of an equal volume of pure, dry air, the temperature and pressure conditions, of course, being the same. The weight of a cubic foot of any gas is designated by δ, and, if the conditions are standard (14.7° pressure per sq. in. and 32° F.), is called the ***specific weight*** δ_0. Since one cubic foot of dry air under these conditions weighs 0.08071 pound, we have the following relation between specific weight and density.

$$\delta_0 = 0.08071\ \Delta. \tag{6}$$

The specific volume, v, of any gas, that is, the number of cubic feet per pound, is, of course, $= \frac{1}{\delta}$, and if the conditions are standard

$$v_0 = \frac{1}{\delta_0}. \tag{7}$$

The table below gives the values of δ_0 and v_0 for a series of gases, but they may also be computed from the following relations. From the general gas law $Pv = RT$, we have

$$v = \frac{1}{\delta} = \frac{RT}{P}. \tag{8}$$

Now for standard conditions, $T = 459.6 + 32 = 491.6°$, and $P = 2117$ pounds per square foot. Hence

$$v_0 = \frac{1}{\delta_0} = \frac{492\,R}{2117} = \frac{R}{4.306}\ \text{cu. ft.} \tag{9}$$

We may also derive from Avogadro's law that

$$\delta_0 = \frac{m}{358}\ \text{lbs. per cu. ft.} \tag{10}$$

where m = the molecular weight of the gas.

Example. — The molecular weight of CO_2 = 44. Hence

$$\delta_0 = \frac{44}{358} = .1229\ \text{lb. per cu. ft.,}$$

and $v_0 = \frac{1}{\delta_0} = \frac{1}{.1229} = 8.14$ cubic feet. Or, since $R = 34.89$ for this gas (see table below), we will also have

$$v_0 = \frac{R}{4.306} = \frac{34.89}{4.306} = 8.10\ \text{cu. ft.}$$

The agreement is close enough for all practical purposes.

Since the volume of any given weight of gas is a function of both pressure and temperature, all gas volumes should be reduced to some standard set of conditions, in order to obtain a comparable basis. The conditions usually assumed are 14.7 pounds pressure per square inch and 32° F., although in view of the fact that 32° is an unusual temperature, some authorities have proposed to use 60° F. Volume varies inversely with absolute pressure and directly with absolute temperature. Consequently if V is the volume of gas under an absolute pressure of P pounds per square inch and an absolute temperature of T degrees F., the volume V_0 under standard conditions will be

$$V_0 = V \cdot \frac{P}{14.7} \cdot \frac{460 + 32}{T} = 33.48 \frac{P}{T} V \text{ cu. ft.} \qquad (11)$$

Where engine guarantees are based upon cubic feet of gas used, the matter of reaching an understanding as to pressure and temperature is of importance. The latter evidently control the *weight* of the number of cubic feet of gas guaranteed. Engine performance is a function of charge weight, not charge volume, and of heat content of the charge, and these factors are largely controlled by pressure and temperature.

Gas.	Atomic Weight.	Molecular Weight, Exact.	Molecular Weight, Approximate.*	Molecular Formula.	Density (Air = 1) Δ	Weight per Cu. Ft. (32° F. and 14.7 Lbs.) δ_0	Specific Volume v_0	Constant R in $Pv = RT$.
Hydrogen	1.008	2.015	2.0	H_2	0.0695	0.00561	178.3	707.7
Oxygen	16.00	16.00	16.0	O_2	1.105	0.0892	11.21	48.27
Nitrogen	14.01	...	28.0†	N_2	0.970†	0.0783†	12.77†	54.94†
Carbon monoxide	...	28.00	...	CO	0.967	0.0781	12.81	55.16
Carbon dioxide	..	44.00	44.00	CO_2	1.529	0.1234	8.103	34.89
Dry air	......	...	29.0‡	..	1.000	0.0807	12.39	53.35
Water vapor	......	18.02	18.0	H_2O	0.622	0.0502	19.94	85.86
Acetylene	......	26.03	26.0	C_2H_2	0.898	0.0725	13.79	59.38
Methane	......	16.03	16.0	CH_4	0.553	0.0446	22.40	96.45
Ethylene	......	28.04	28.0	C_2H_4	0.938	0.0781	12.81	55.14

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* Sufficiently accurate for most engineering calculations.

† These figures are for the so-called "nitrogen" of the atmosphere, which carries about 0.5 per cent by volume of the heavy inert gas "argon."

‡ An equivalent value often useful in computations. For carbon, if that can be imagined to exist as a gas under standard conditions: Atomic weight = 12, Δ = .820, δ_0 = .0668, and v_0 = 14.97.

(*b*) ***Constants for the Fuel Gases, Heating Values, Air Required for Combustion, etc.*** — Combustible gases, like CO, H, or the hydrocarbons, are hardly ever used singly in gas-engine mixtures, the commercial fuel gases being nearly always composite. The constants for the single gases are, however, necessary for the computations of heating value, and air required for combustion, and these are therefore given in the following table. The heating values given in this table are calorimetric, not computed. A computation does not take into account the heat rendered latent in the breaking up of the hydrocarbon combination, hence the computed results are uniformly too high. The error may be as high as 12 per cent, although there are empirical formulas which come very close.

Gas.	Density. Air = 1.0.	Higher Heating Value, B.t.u. per Lb.	Higher Heating Value, B.t.u. per Standard Cu Ft	Lower Heating Value, B.t.u. per Lb.*	Theoretical Amount of			Lbs. Products of Comb. per Lb. of Gas.		
					Oxygen per Lb.	Air for		CO₂	H₂O	N
					Lbs	1 Lb. Lbs.	1 Cu. Ft Cu. Ft.			
H_2	.0695	61,950	345	52,500	8	34.78	2.40	. . .	9	26.78
CO	.9670	4,380	342	4,380	.57	2.48	2.38	1.57	. . . .	1.91
CH_4	5540	23,840	1,067	21,380	3 90	17 30	9.56	2 74	2.25	13 31
C_2H_2	.9150	21,430	1,582	20,670	3 07	13 35	11 99	3 38	69	10 28
C_2H_4	9740	21,430	1,685	20,020	3 43	14.95	14.58	3.14	1.29	11 52
C_2H_6	1.0367	22,400	1,873	20,430	3 72	16 17	16 76	2.92	1 80	12 45
C_3H_4	1.3819	20,990	2,342	20,010	3.19	13.87	19.15	3.29	90	10 68
C_3H_6	1.4512	21,220	2,490	19,820	3 42	14 87	21.55	3 14	1 28	11 45
C_3H_8	1 5204	21,830	2,675	20,040	3.63	15.78	23.98	2 99	1.64	12.15
C_4H_8	1.9349	20,910	3,275	19,510	3 43	14 95	28 86	3.14	1 29	11 52

* For a discussion of the correction to be applied to the higher heating value in case the water vapor formed escapes as steam, see Chap XIII, p 467.

With the aid of the constants in the above table, it is possible to compute the characteristics of a commercial gas, such as illuminating or producer gas. To simplify the computation, a formula for the air required for combustion may be established. This may be based either on the pound or on the standard cubic foot of the gas.

Let *one pound* of the gas consist of

$$x_1 \text{ lbs. CO} + x_2 \text{ lbs. } H_2 + x_3 \text{ lbs. } CH_4 + x_4 \text{ lbs. } C_2H_4 + x_5 \text{ lbs. } C_2H_2$$
$$+ x_6 \text{ lbs. } O_2 + x_7 \text{ lbs. } N_2 + x_8 \text{ lbs. } CO_2 + x_9 \text{ lbs. } H_2O.$$

Then theoretical air required for combustion

$$= \frac{.57 x_1 + 8 x_2 + 3.99 x_3 + 3.43 x_4 + 3.07 x_5 - x_6}{.23} \text{ lbs. per lb.} \quad (12)$$

The products of combustion for the theoretical air supply will be

$$CO_2 = [1.57 x_1 + 2.74 x_3 + 3.14 x_4 + 3.38 x_5 + x_8] \text{ lbs.} \quad (13)$$

$$H_2O = [9 x_2 + 2.25 x_3 + 1.29 x_4 + .69 x_5 + x_9] \text{ lbs.} \quad (14)$$

$$N_2 = [.77 \text{ (wt. of theoretical air from eq. (12))} + x_7] \text{ lbs.} \quad (15)$$

Also let *one cubic foot* of the gas consist of

x_1 cu. ft. CO + x_2 cu. ft. H_2 + x_3 cu. ft. CH_4 + x_4 cu. ft. C_2H_4 + x_5 cu. ft. C_2H_2 + x_6 cu. ft. O_2 + x_7 cu. ft. N_2 + x_8 cu. ft. CO_2 + x_9 cu. ft. H_2O.

Theoretical air required for combustion

$$= \frac{\frac{x_1 + x_2}{2} + 2 x_3 + 3 x_4 + 2.5 x_5 - x_6}{.21} \text{ cu. ft. per cu. ft.} \quad (12a)$$

Products of combustion per cubic foot of gas for the theoretical air supply will be

$$CO_2 = [x_1 + x_3 + 2 x_4 + 2 x_5 + x_8] \text{ cu. ft.} \quad (13a)$$

$$H_2O = [x_2 + 2 x_3 + 2 x_4 + x_5 + x_9] \text{ cu. ft.} \quad (14a)$$

$$N_2 = [.79 \text{ (cu. ft. of theoretical air from eq. (12a))} + x_7] \text{ cu. ft.} \quad (15a)$$

In either case, if the ratio of air to gas is greater than the theoretical, as it usually is, the products of combustion will show in addition a certain quantity of excess air. This may be considered to consist of nitrogen and free oxygen. The former is added to that found under equations (15) or (15a), so that the products of combustion will show CO_2, H_2O, N_2, and O_2, provided, of course, that the combustion is complete.

Example. — An illuminating gas shows the following composition by volume: 48.50 per cent H_2, 35.00 per cent CH_4, 7.00 per cent CO, heavy hydrocarbons (considered as C_2H_4) 4.50 per cent, 2.00 per cent CO_2, .25 per cent O_2, 2.75 per cent N_2. Assume that the air used for combustion is 50 per cent in excess of that required, i.e., that the excess coefficient is 1.5.

Higher heating value, per cubic foot,

$$= [(345 \times .485) + (1067 \times .35) + (342 \times .07) + (1685 \times .045)] = 640 \text{ B.t.u.}$$

Theoretical air required for combustion, per cubic foot,

$$= \frac{\left[\frac{.07 + .485}{2} + (2 \times .35) + (3 \times .045) - .0025\right]}{.21} = 5.28 \text{ cu. ft.}$$

Air actually supplied = 1.5 × 5.28 = 7.92 cubic feet per cubic foot of gas.

Products of combustion:

$CO_2 = [.07 + .35 + 2 \times .045 + .02] = .530$ cu. ft.
$H_2O = [.485 + 2 \times .35 + 2 \times .045] = 1.275$ cu. ft.
$N_2 = [.79 \times 5.28 + .0275] + .79\,(7.92 - 5.28) = 6.285$ cu. ft.
$O_2 = .21\,(7.92 - 5.28) = .554$ cu. ft.

The sum total of the volumes of the products of combustion is 8.64 cubic feet. The original volume of the mixture was 1 + 7.92 = 8.92 cubic feet. Hence there has been a contraction during combustion amounting to

$$\frac{8.92 - 8.64}{8.92} = 3.1 \text{ per cent,}$$

assuming that the products are brought back to initial pressure and temperature. The exhaust gases will show the following composition by volume: 6.13 per cent CO_2, 14.75 per cent H_2O, 72.71 per cent N_2, and 6.41 per cent free O_2. In our ordinary method of analysis, the greater part of the water vapor is condensed, so that the analysis of these exhaust gases as made will not agree with the above figures. See Chap. XIII.

Similar computations may be made for any of the commercial gases. The table, p. 861, shows average composition and average constants for the most important of these gases. The figures may be used as a check upon computations made in practice.

(*c*) *Characteristics of the Liquid Fuels.* – For the liquid fuels, the computations may be carried through the same way, once the composition of the oils and the ratio of air to oil are known. Crude oil and its distillates, kerosene and gasoline, show in general about the same composition, which is not far from 84 to 87 per cent C by weight, 11.5 to 14.5 per cent H, and .5 to 4 per cent of oxygen and impurities. Taking the average at 85 per cent C_2, 14 per cent H_2, and 1 per cent impurities (O_2), the theoretical air required per pound should be

$$\frac{.85 \times 2.66 + .14 \times 8 - .01}{.23} = 14.7 \text{ lbs.}$$

Av. Comp Per Cent by Vol	Illuminating Gas.	Oil Gas.*	Anthracite Producer Gas.†	Bituminous Coal Producer Gas.†	Blast Furnace Gas.	Coke Oven Gas.	Natural Gas.‡	Water Gas.
CO	8.18	8.9	27.0	27.0	26.0	7.0	.73	45.0
H_2	46.20	5.6	12.0	12.0	3.0	55.0	1.86	45.0
CH_4	34.00	54.9	1.2	2.5	.5	32.0	93.07	2.0
C_2H_4	3.76	28.9	....	4	...	1.5	47	.....
CO_2	8.88	9	2.5	2.5	9.5	1.2	.26	4.0
N_2	2.15	..	57.0	56.2	56.0	1.5	3.04	2.0
O_2	.65	...	.3	.3	...	...	.42	.5
H_2O	1.50	....	....	......	5.0	1.0	..	......
Specific weight, lbs. per cu. ft.	.032	.058	.065	.065	.079	.027	.046	.045
Higher heating value, B.t.u. per cu. ft.	613	1,120	147	168	10	580	1,010	330
per lb.	19,150	19,300	2,260	2,590	1,330	21,500	22,000	7,350
Minimum air required for comb. cu. feet per cu. foot	5.25	9.5	1.00	1.15	7	5.0	9.0	2.3

* Made by vaporizing crude oils. Should be distinguished from the water-oil gas made by the Lowe process.

† These analyses are from an R. D. Wood catalogue. The composition of producer gas may vary over wide ranges; thus Güldner gives the following for an anthracite gas: 16.6 per cent CO; 24.2 per cent H_2; 2 per cent CH_4, 11.3 per cent CO_2; 45.9 per cent N_2.

‡ Anderson, Indiana

The products of combustion for the same oil, burned with the theoretical air supply, will be .85 × 3.66 = 3.11 pounds CO_2; .14 × 9 = 1.26 pounds H_2O; and .77 × 14.7 = 11.32 pounds N_2.

The heating value of these oils is fairly constant, as might be expected from the constancy of the composition, the range being from about 17,500 to 21,000 B.t.u. per pound.

The only other liquid fuel of any importance is ethyl alcohol, the chemical formula for absolute alcohol being C_2H_6O. This composition shows .522 pound C_2, .130 pound H_2, and .348 pound O_2 per pound of liquid. The theoretical air required for combustion is 9 pounds per pound. Commercial alcohol, however, always carries some water, so that specific gravity, heating value, air required for combustion, etc., all change with this variable factor. The proportion of absolute alcohol in a mixture of alcohol and water is expressed as a percentage by volume or by weight. The following table shows these figures for various admixtures of water.

Absolute Alcohol		Specific Gravity at 59° F	Higher Heating Value, per Lb B t u.
Volume, Per Cent	Weight, Per Cent		
95	93 8	805	12,140
90	87 7	815	11,340
85	81 8	826	10,580
80	76 1	836	9,850
75	70 5	846	9,120

The vapor volumes resulting when any of the above liquid fuels are vaporized are, of course, a function of pressure and temperature. For information on this point, the student is referred to books on the subject of gas engines.

(*d*) *Specific Heats.* - The specific heat of the gases and vapors is one of the factors necessary to the computation of heat changes accompanying temperature changes. Specific heats are a function of both pressure and temperature. The variation with pressure, as far as the gases commonly encountered are concerned, seems to be minor, *except for water vapor*, but the change in the value of the specific heat with temperature changes is of sufficient degree to compel recognition where accuracy is desired. Although a great deal of experimental work has been done in this field, the results of the different investigations are not as yet in entire accord.

The following data are an abstract of an investigation made on the subject by Prof. G. B. Upton, in which he collaborated the results of the most important experiments made by Mallard and Le Chatelier, Holborn and Henning, Langen, Pier, and others. The aim in view was not to attempt to reconcile the various results obtained, but by a judicious balancing of all the facts, to furnish data on specific heats which would be generally useful in gas computations, with a degree of accuracy, on the basis of the present state of our knowledge, quite sufficient for engineering work. As a result, the equations furnished by one experimenter were accepted for one gas, while those of another were used for another gas. Thus Pier's, and Holborn and Henning's results were taken for oxygen and carbon dioxide, nitrogen and carbon monoxide, Langen's results for hydrogen. The data for water vapor are the result of a combination of available figures, mainly Holborn and Henning's.

This is the only constituent of commercial gases in which the pressure plays any important part as far as specific heat is concerned. *Now in the case of flue or exhaust gases, the partial pressure of the vapor, which is the pressure criterion for the determination of specific heat, is rarely over 1 pound absolute, while at the explosion pressure in a gas engine it rarely exceeds 50 pounds absolute.* At the same time the low pressure range (i.e., about 1 pound absolute) is accompanied by temperatures not very much exceeding 400° C. (752° F.), and serviceable data for this range may therefore be obtained from Marks and Davis' Steam Tables for 1 pound pressure. The higher pressure range, on the other hand, is accompanied by high temperatures (up to 3500° F.). In that temperature range the field of variation of specific heats of water vapor with pressure narrows very rapidly (note the indicated narrowing of the field beyond 400° F. in Fig. 244), so that there is little practical difference between the specific heats at 50 pounds absolute, and those at atmospheric pressure. For the latter we have Holborn and Henning's results and these may consequently be accepted for gas computations made at any of the pressures occurring in the gas engine cycle. The curve (in Fig. 537) expressing the variation of specific heat with temperature in the case of water vapor is therefore constructed up to 400° C. with the data for 1 pound absolute pressure from Marx and Davis' Steam Tables, while beyond that Holborn and Henning's results for 15 pounds absolute are used. The equation given below for H_2O closely represents this curve. This eliminates the pressure function also for H_2O in the case of any combustion computations that are likely to be made in experimental engineering work.

Definitions of specific heat at constant pressure, C_p, and at constant volume, C_v, have already been given in Art. 174.

Under each, we distinguish further two kinds of specific heat, the mean and the instantaneous. As the name indicates, the *mean specific heat* is the value by which a temperature range must be multiplied to obtain the quantity of heat which was required to raise unit weight of the material through the range stated, under the conditions obtaining. The *instantaneous specific heat* is the quantity of heat that must be supplied to unit weight of a material to raise the temperature *one degree* under stated conditions of pressure and volume. The mean specific heats are, of course, used for all heat calculations, while the instantaneous values must be used for the determination of $\gamma = \frac{C_p}{C_v}$. The notation used below is as follows: Instantaneous specific heat at constant pressure $= C_{pi}$; at constant volume $= C_{vi}$; mean specific heat at constant pressure $= C_{pm}$; at constant volume $= C_{vm}$. We also speak of *molecular specific heats*, obtained by multiplying each of the specific heats above mentioned by the molecular weight of the gas, m; thus mean molecular specific heat at constant pressure $= m \cdot C_{pm}$, etc. This quantity is little used in engineering work.

Practically all of the experimental work in connection with specific heats has been done with the Centigrade scale of temperature. Hence this scale is retained in all the equations, the transposition being made only in the curves.

It can be shown that if C_i, the instantaneous specific heat at any temperature, t, is represented by the function

$$C_{i\,(t)} = a + bt + ct^2 + dt^3 + et^4 + \qquad (16)$$

the mean specific heat in the range from 0 to the same temperature, t, can be represented by

$$C_{m\,(0-t)} = a + \frac{b}{2}t + \frac{c}{3}t^2 + \frac{d}{4}t^3 + \frac{e}{5}t^4. \qquad (17)$$

These relations will serve for the conversion of mean into instantaneous specific heats or *vice versa*. Note that the mean specific heats are computed above 0° C.

Concerning the inter-relation of C_{pi} and C_{vi}, Boynton has shown that for N_2, O_2, H_2, CO, CO_2, H_2O, SO_2, and NH_3, we may write

$$\tfrac{3}{2}\, mC_{vi}\,(\gamma - 1) = 3; \quad \text{or} \quad mC_{vi}\,(\gamma - 1) = 2.00. \qquad (18)$$

Since $\gamma = \frac{C_{pi}}{C_{vi}}$, this expression may also be written

$$m\,(C_{pi} - C_{vi}) = 2; \quad \text{or} \quad C_{pi} - C_{vi} = \frac{2}{m}. \qquad (19)$$

How closely this equation is actually fulfilled, as judged from the basis of existing experimental results, may be seen from the following:

For	CO	Air	H_2O	H_2
$mC_{pi} - mC_{vi} =$	2.017	1.960	2.020	1.955

The theoretical value for γ is the same for all monatomic gases (= 1.667); for all diatomic gases (=1.400); for all triatomic gases (= 1.286) etc., assuming no association or dissociation to occur. Consequently C_{pi} and C_{vi} should be the same for all gases of the same atom number in the molecule. Experimental data show that this condition is not quite met (see the curves below), but how far this is due to errors involved in the investigations is, of course, problematical.

The following equations represent the final results of Professor Upton's investigation, except for the case of H_2, for which Langen's equation is used. (Temperatures are in degrees C.)

Oxygen (O_2)	Mean specific heat	$C_{pm} = 0.216 + 0.0000014\,t$ $C_{vm} = 0.153 + 0.0000014\,t$
	Instantaneous specific heat	$C_{pi} = 0.216 + 0.0000028\,t$ $C_{vi} = 0.153 + 0.0000028\,t$
Nitrogen (N_2) and Carbon monoxide (CO)	Mean specific heat	$C_{pm} = 0.243 + 0.000019\,t$ $C_{vm} = 0.171 + 0.000019\,t$
	Instantaneous specific heat	$C_{pi} = 0.243 + 0.000038\,t$ $C_{vi} = 0.171 + 0.000038\,t$
Air	Mean specific heat	$C_{pm} = 0.237 + 0.000019\,t$ $C_{vm} = 0.168 + 0.000019\,t$
	Instantaneous specific heat	$C_{pi} = 0.237 + 0.000038\,t$ $C_{vi} = 0.168 + 0.000038\,t$
Carbon dioxide (CO_2)	Mean sp. ht.	$C_{pm} = .200 + 75\times10^{-6}\,t - 21\times10^{-9}\,t^2 + 2.2\times10^{-12}\,t^3$ $C_{vm} = .155 + 75\times10^{-6}\,t - 21\times10^{-9}\,t^2 + 2.2\times10^{-12}\,t^3$
	Inst. sp. ht.	$C_{pi} = .200 + 150\times10^{-6}\,t - 65\times10^{-9}\,t^2 + 9.1\times10^{-12}\,t^3$ $C_{vi} = .155 + 150\times10^{-6}\,t - 65\times10^{-9}\,t^2 + 9.1\times10^{-12}\,t^3$
Water vapor (H_2O)	Mean sp. ht.	$C_{pm} = .452 + 7.4\times10^{-6}t + 92.6\times10^{-9}t^2 - 20.6\times10^{-12}t^3$ $C_{vm} = .340 + 7.4\times10^{-6}t + 92.6\times10^{-9}t^2 - 20.6\times10^{-12}t^3$
	Inst. sp. ht.	$C_{pi} = .452 + 14.8\times10^{-6}t + 278\times10^{-9}t^2 - 82\times10^{-12}t^3$ $C_{vi} = .340 + 14.8\times10^{-6}t + 278\times10^{-9}t^2 - 82\times10^{-12}t^3$.

For hydrogen (H_2) the following may be used:

Hydrogen (H_2)	Mean sp. heat	$C_{pm} = 3.369 + 0.00055\,t$ $C_{vm} = 2.369 + 0.00055\,t$
	Inst. sp. heat	$C_{pi} = 3.369 + 0.0011\,t$ $C_{vi} = 2.369 + 0.0011\,t$

The information on CH_4 with regard to specific heats is very meager. The writer has been able to find only one equation, that given by Richards in his Metallurgical Calculations, Part I, p. 110 The equation there given, computed to the ordinary basis, shows the following:

$$\text{Methane } (CH_4) \begin{cases} \text{Mean sp. heat} & \begin{cases} C_{pm} = .532 + .0003\,t \\ C_{vm} = .407 + .0003\,t \end{cases} \\ \text{Inst. sp. heat} & \begin{cases} C_{pt} = .532 + .0006\,t \\ C_{vt} = .407 + .0006\,t \end{cases} \end{cases}$$

To save the labor of solving these equations for any ordinary case, the curve sheet, Fig 537, gives a graphical solution of them up

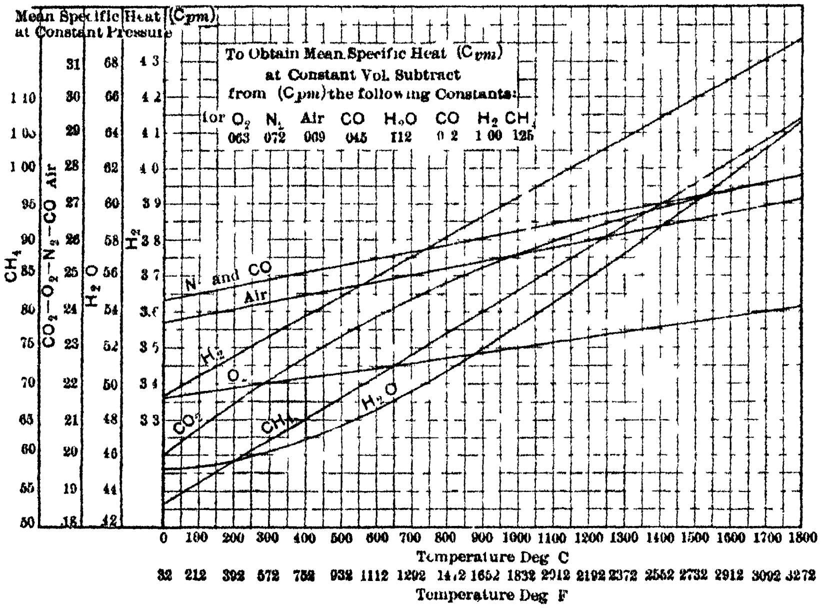

FIG 537

to 1800° C. (3272° F) It should be stated that beyond 1000° C the results are somewhat uncertain and that in consequence dependence can be placed in only the first two significant figures in the decimal. The curves give the mean specific heat at constant pressure (C_{pm}) between 0° C. (or 32° F) and any other given temperature t From this the value of C_{vm} may be derived, as shown on the sheet.

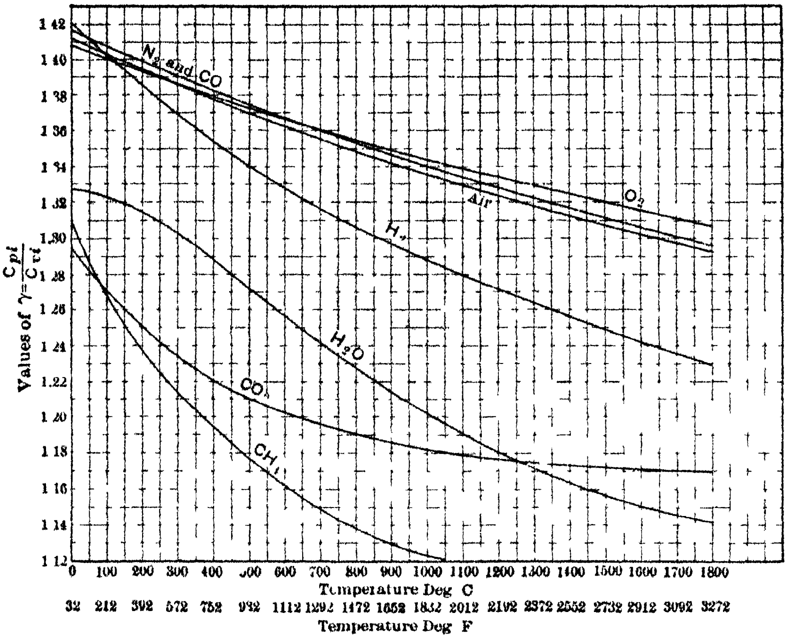

FIG 538

The second curve sheet, Fig. 538, gives values of $\gamma = \frac{C_{pt}}{C_{vt}}$, as the change in this figure is large enough to be of importance in some cases.

It happens quite often that the quantity of heat gained or lost between some temperature t' (not 0° C or 32° F) and a temperature t is desired This case occurs, for instance, when it is desired to compute the heat loss in a gas from a temperature of t degrees to room temperature at t' degrees. The most obvious way of solving this problem is to compute the heat in the gas at t degrees by use of $C_{pm\ (0\ \text{to}\ t)}$ and to subtract from this the heat still in the gas at t' degrees by use of $C_{pm\ (0\ \text{to}\ t')}$ But it can be shown mathematically that if mean specific heat can be expressed by an equation of the general form

$$C_{pm\ (0\ \text{to}\ t)} = x + yt, \quad \ldots\ldots\ldots \quad (20)$$

then

$$C_{pm\ (t'\ \text{to}\ t)} = x + y\,(t' + t) \quad \ldots\ldots \quad (21)$$

The use of the last equation, of course, shortens the work.

Example. — Suppose that *one pound* of air is cooled from 2000° F. to 60° F. What is the heat loss?

Method 1. C_{pm} at 2000° = .258; at 60° = .237, from the curves.

$$\begin{aligned}\text{Heat in air above } 32° \text{ at } 2000° &= .258\,(2000 - 32) = 507.7 \text{ B.t.u.}\\ \text{Heat in air above } 32° \text{ at } 60° &= .237\,(60 - 32) = 6.6 \text{ B.t.u.}\\ \text{Heat loss} &= 501.1 \text{ B.t.u.}\end{aligned}$$

Method 2. Equation for C_{pm} for air

$$= C_{pm} = .237 + .000019\,t,$$

in which t = centigrade degrees.

Changing this to the form of equation (21), with t' = 15.5° C. (60° F.) and t = 1093° C. (2000° F.), we have

$$C_{pm\,(60 \text{ to } 2000)} = .237 + .00019\,(15.5 + 1093) = .258$$

$$\text{Heat loss} = .258\,(2000 - 60) = 500.5 \text{ B.t.u.}$$

386. Analysis of Fuel Gases and the Determination of Heating Value. — The chemical analysis of fuel gases is a matter requiring care and experience, and unless both of these requirements can be fulfilled, it will in general be best, if the plant test is made in the field, to collect the gas by some approved method and to send it to some chemical laboratory for analysis.

Means for collecting gas are discussed in Chapter XIII, to which the student is referred. Where, in any given plant, the sample of gas should be taken, depends upon the purpose in view. If the test is confined to the engine, the gas should, of course, be collected close to the engine, no attention being paid to the previous history of the gas. In case a producer is under test, it is always well to sample the gas just at the outlet of the producer, and again after leaving the cleaning apparatus on its way to gas holder or to engine. Where a test is made of the plant complete, the two analyses just mentioned are sufficient to cover the requirements. For remarks concerning number of samples, method of averaging the results of the various analyses, etc., see p. 876.

The analysis consists in the determination of CO_2, O_2, C_2H_4, CO, CH_4, and H_2, N_2 being found by difference. It may be carried out in two ways:

(*a*) To determine CO_2, O_2, C_2H_4, and CO by absorption; H_2 by catalytic separation by means of palladium; and CH_4 by combustion.

(*b*) To determine only CO_2, O_2, and C_2H_4 by absorption, H_2, CO, and CH_4 by combustion.

As far as the work in a chemical laboratory is concerned, there is probably little choice between these two methods, but in field work method (*b*) offers decided advantages, as the apparatus is simpler and requires less time to operate. A distinction should be made between gases based upon the CO content. Where this is comparatively low, as in illuminating gas, an absorption method, like (*a*), gives satisfactory results; but for producer gas or water gas, both high in CO, method (*b*) would be preferred, especially in the field, on account of the difficulty of absorbing large quantities of CO without thorough agitation of gas and reagent.

The reagent used for the absorption of CO_2 is caustic potash, that for O_2 is either an alkaline solution of pyrogallic acid or phosphorus. The preparation and handling of these reagents has already been described in Chapter XIII. The reagent used for C_2H_4 is fuming sulphuric acid. The remainder of the constituents, except nitrogen, are combustible. They are transferred to a combustion tube or vessel and mixed with a measured quantity of air. The mixture is lighted by means of incandescent platinum. After combustion, the resulting CO_2 and the free oxygen remaining in the gases are determined by absorption. The reactions occurring in the combustion tube may be written as follows:*

$$\begin{aligned} 2\,CO + O_2 &= 2\,CO_2, \\ \text{i.e., } 2 \text{ vols. of } CO + 1 \text{ vol. } O_2 &= 2 \text{ vols. of } CO_2. \quad \ldots \quad (22) \end{aligned}$$

$$\begin{aligned} CH_4 + 2\,O_2 &= CO_2 + 2\,H_2O, \\ \text{i.e., } 1 \text{ vol. } CH_4 + 2 \text{ vols. } O_2 &= 1 \text{ vol. } CO_2 + 2 \text{ vols. } H_2O. \quad . \quad (23) \end{aligned}$$

$$\begin{aligned} 2\,H_2 + O_2 &= 2\,H_2O, \\ \text{i.e., } 2 \text{ vols. } H_2 + 1 \text{ vol. } O_2 &= 2 \text{ vols. } H_2O. \quad \ldots\ldots \quad (24) \end{aligned}$$

With the aid of these expressions we may derive the following equations:

---

* This development of the equations used for the determination of CO, H_2 and CH_4 after combustion was apparently first given by Vignon, Bull. Soc. Chim. 1897, 832 It is contained in Hempel-Dennis' Gas Analysis.

Let Σ = the sum of the contraction volumes appearing in equations (22) to (24).
V = the sum of the volumes of CO, CH_4, and H_2,
and CO_2 = the sum of the CO_2 volumes produced in equations (22) and (23).

In equation (22), 3 volumes before combustion contract to 2 volumes; hence the contraction amounts to 1 volume, which is $\frac{1}{2}$ of the CO volume concerned in the equation. The contraction volume may therefore be represented by $\frac{1}{2}$ CO.

In equation (23), 3 volumes before combustion contract to 1 volume, since the 2 volumes of H_2O produced will condense. Hence the contraction volume is equal to 2 volumes, which is twice the volume of CH_4 concerned in the equation. The contraction volume is therefore represented by 2 CH_4.

In equation (24), 3 volumes before combustion contract to 0 volume, since the 2 volumes of H_2O will condense. Hence the contraction volume is equal to 3 volumes, which is $\frac{3}{2}$ of the volume of H_2 concerned in the equation. The contraction volume may therefore be represented by $\frac{3}{2}$ H_2.

Equation (22) shows that the volume of CO_2 produced is the same as the volume of CO concerned in the equation, while equation (23) shows the same to hold true for the relation between the volumes of CO_2 and CH_4.

With this information we now have

$$\Sigma = \frac{1}{2}\,CO + 2\,CH_4 + \frac{3}{2}\,H_2, \qquad (25)$$

$$V = CO + CH_4 + H_2, \qquad (26)$$

$$CO_2 = CO + CH_4. \qquad (27)$$

Solving these for each one of the combustible gases, we finally derive

$$CO = \frac{1}{3}\,CO_2 + V - \frac{2}{3}\,\Sigma, \qquad (28)$$

$$H_2 = V - CO_2, \qquad (29)$$

$$CH_4 = \frac{2}{3}\,CO_2 - V + \frac{2}{3}\,\Sigma. \qquad (30)$$

How these equations are practically applied will be shown by a type example.

There are several types of apparatus, based generally upon the principles of the Orsat, which should be satisfactory for field work. One of the best is probably the Hahn apparatus* described by Dr. C. Hahn in the Zeitschrift des Vereins deutscher Ingenieure for Feb. 10, 1906. Another is described in the same journal for April, 1908, and is manufactured by Dr. Siebert & Kuhn, Kassel.†

The general appearance of the Heinz apparatus is shown in Fig. 539, while Fig. 540 shows in detail the arrangement of the parts.

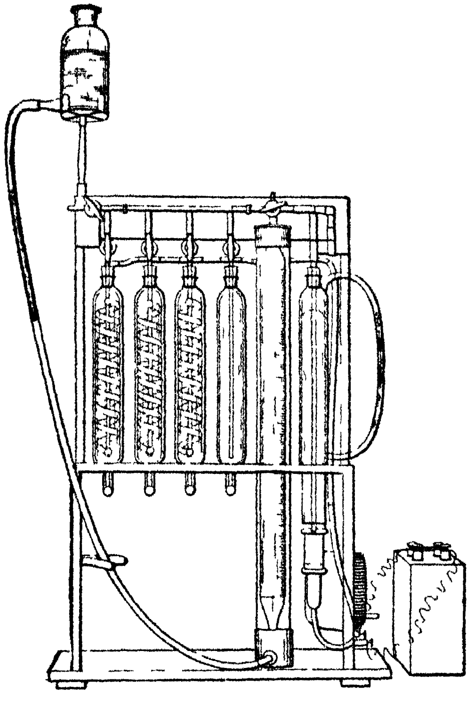

FIG 539 —HEINZ APPARATUS FOR GAS ANALYSIS

a, *b*, *c* and *d* are the absorption pipettes, *e* is the measuring burette, and *f* the combustion pipette. The latest improved forms of absorption pipettes are shown in Figs. 541 and 542, the former of which shows the Hankus, the second the Heinz pipette. Both are designed to promote thorough intermixing of gas and reagent, and the rapidity and certainty of action are very much better than in the old type. In the Hankus pipette, gas enters at *a* and flows out

---

* Made by C Heinz, Aachen

† Another apparatus is described by Paul Fuchs in his "Generator Kraftgas und Dampf-Kessel Betrieb." It is made by G. A. Schultze, Charlottenburg.

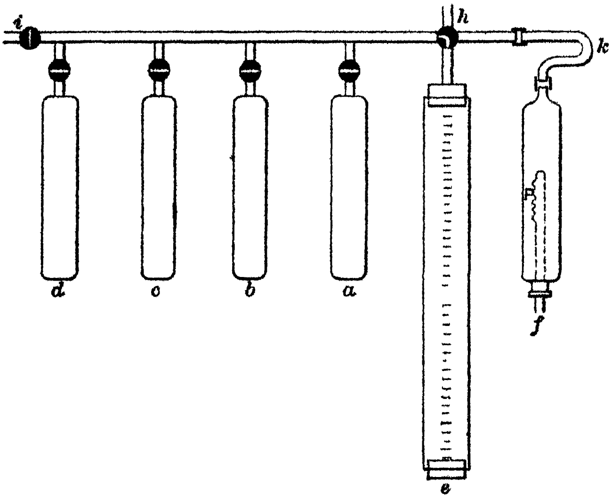

FIG. 540.—ARRANGEMENT OF PARTS OF HEINZ APPARATUS FOR GAS ANALYSIS

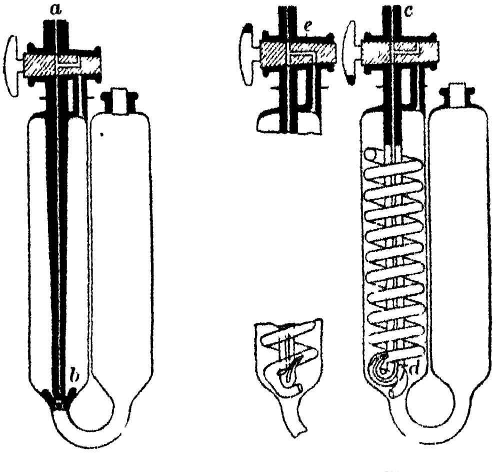

FIG. 541.—ABSORPTION PIPETTES—FIG. 542.

at *b* into the reagent, through which it rises to the top. It may then be drawn out through the side opening at the top by throwing the cock over into the right position (shown at *e* in the detail drawing for the Heinz pipette). The action of the Heinz pipette is very similar. The gas flows from *c* to *d* through the central tube, escapes through a small injector opening and rises in the long spiral. It is removed as in the Hankus pipette.

The analysis is carried out as follows:

Assuming that the liquids in the absorption and combustion pipettes (mercury or water in the latter) stand at the top of the capillary tube, that the measuring burette and capillary are filled with the displacing liquid (mercury or water), a sample of gas is drawn into the apparatus through the cock *i*. After measuring in *e*, the gas is displaced into *a*, which contains *KOH* for the absorption of CO_2. After this is complete and the contraction has been measured, displace into *b*, which contains fuming sulphuric acid for the absorption of the heavy hydrocarbons (considered as C_2H_4). From *b* draw the gas back into *e*, but before determining the contraction, displace over into *a* again to take out acid fumes. Then draw back into *e* and measure the total contraction which determines $CO_2 + C_2H_4$. Next displace into *c*, containing pyro-gallol for the absorption of O_2. After absorption in *c*, the total contraction shown in *e* measures $CO_2 + C_2H_4 + O_2$. It is next possible to absorb CO by means of a cuprous chloride solution in the pipette *d*. But owing to the fact that producer gases contain a good deal of CO (up to 30 per cent in some cases), and that the absorbing power of the solution for CO is very limited, the writer prefers to omit this absorption and to determine CO by combustion, as above stated. To this end, have the pipette *d* simply filled with water, and at the end of the O_2 absorption displace all the gas over into *d*, which may then serve as a storage. Next draw into *e* a certain quantity of air through *h*, measure it and force it over into the combustion pipette *f*. Then draw over from *d* a certain quantity of gas and measure it. The proportion of air to gas will depend upon the kind of gas; more than enough oxygen to completely burn all of the combustible components (CO, H_2 and CH_4) must be present. Next pass a current through the platinum spiral *P*, by means of the storage battery shown in Fig. 539, so that this wire will be at a bright red heat. Then slowly displace the gas from *e* into *f*. The combustion taking place will burn CO to CO_2, H to H_2O and CH_4 to CO_2 and H_2O. After combustion transfer the mixture of burned gases first to pipette *a* to absorb the CO_2 formed by the combustion; measure the contraction in *e* Then transfer to pipette *c* to absorb excess oxygen and again measure the contraction. The analysis is then complete.

The apparatus described by Dr. Hahn differs slightly from this in that there is a palladium wire placed in the bend *k*, Fig. 540, which wire is heated by means of an external flame to a temperature of 400°-500°C. With the spiral *P* cold, the mixture of gas and air is forced from *e* over into *f*, when combustion of H_2 alone will take place in *k*. Afterward spiral *P* is used to burn CH_4, Dr. Hahn using the absorption method for CO. This method then determines H_2, CO and CH_4 separately. The separation of H_2 by this method, however, requires considerable care and skill, and

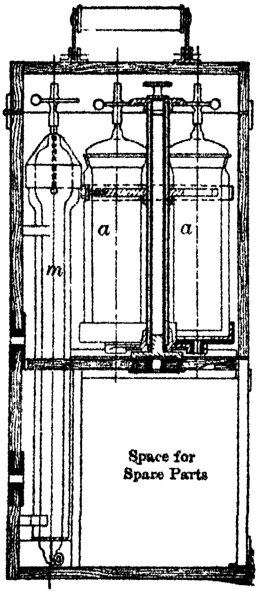

FIG. 543.—SIEBERT-KUHN APPARATUS FOR GAS ANALYSIS.

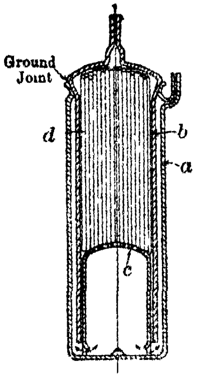

FIG. 544.—ABSORPTION PIPETTE, SIEBERT-KÜHN APPARATUS.

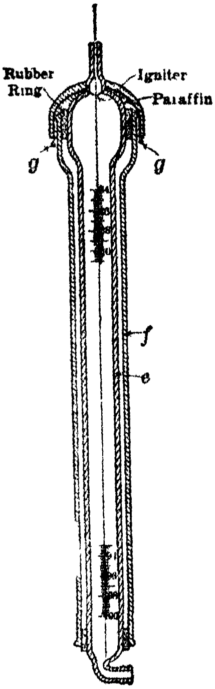

FIG. 545.—MEASURING BURETTE, SIEBERT-KÜHN APPARATUS.

the writer would, therefore, recommend the straight combustion method for CO, H_2 and CH_4 as above outlined.

The Siebert-Kühn apparatus is similar in principle but of different construction. In Fig. 543, *aa* are the absorption pipettes, mounted in a holder that can be revolved on a base, and *m* is the measuring burette. Fig. 544 shows the construction of the absorption pipettes and indicates the means taken for exposing large surfaces of the reagent. The burette, Fig. 545, consists of the graduated tube *e*, surrounded by the jacket tube *f*. The electrodes *gg* lead the current to the platinum spiral in the top of the burette. When ready for use the apparatus is arranged as in Fig. 546. The burette is connected to each absorption pipette in proper order, the combustion part of the analysis being carried on in the burette itself.

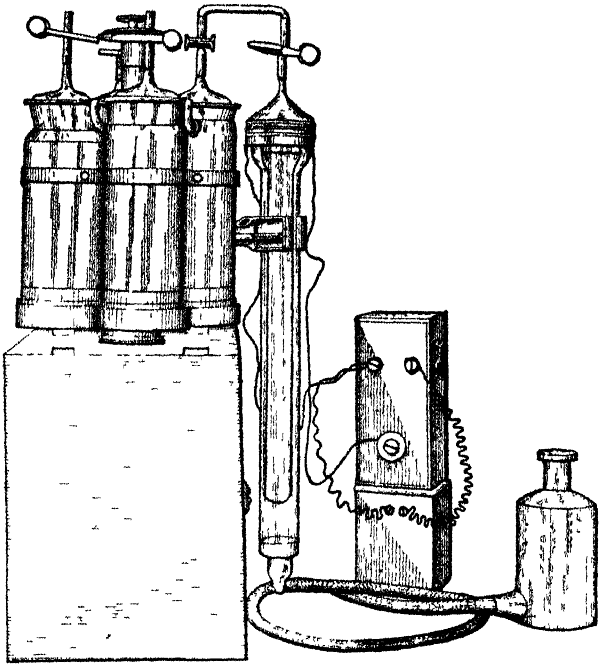

FIG. 546. — SIEBERT-KÜHN APPARATUS READY FOR USE.

Both types of apparatus are put up in compact form in traveling cases.

The following type example of an analysis of a producer gas is taken from the work by Paul Fuchs, mentioned in footnote, p. 871.

Original volume of producer gas	94.8 c.c.	
After absorbing CO_2	90.2 c.c. =	4.6 c.c. CO_2
After absorbing O_2	89.9 c.c. =	.3 c.c. O_2
After absorbing C_2H_4	89.7 c.c. =	.2 c.c. C_2H_4
Volume of gas used for combustion	34.2 c.c.	
Air added	36.7 c.c. =	29.0 c.c. N_2 7.7 c.c. O_2
Volume of air and gas	70.9 c.c.	

Volume after combustion .. 58.3 c.c.; hence $\Sigma = 70.9 - 58.3 = 12.6$ c.c.
After absorbing CO_2 ...... 52.1 c.c.; hence $CO_2 = 58.3 - 52.1 = 6.2$ c.c.
After absorbing O_2 .... 50.9 c.c.
Volume of N_2 added........ 29.0 c.c.
Remaining difference....... $= 21.9$ c.c. = (Volume left of original 34.2 c.c. of gas used) hence $V = 34.2 - 21.9 = 12.3$ c.c.

Substituting these values of Σ, CO_2, and V in equations (23) to (25), we obtain:

$$
\begin{aligned}
CO &= \tfrac{1}{2}\,CO_2 + V - \tfrac{2}{3}\,\Sigma, \\
&= 2.1 + 12.3 - 8.4 = 6.0 \text{ c.c.} \\
H_2 &= V - CO_2, \\
&= 12.3 - 6.2 = 6.1 \text{ c.c.} \\
CH_4 &= \tfrac{2}{3}\,CO_2 - V + \tfrac{2}{3}\,\Sigma, \\
&= 4.1 - 12.3 + 8.4 = .2 \text{ c.c.}
\end{aligned}
$$

The ratio between the original volume of gas taken and that used for combustion is $\frac{94.8}{34.2} = 2.77$. Therefore in the original volume there were contained .6 c.c. of CH_4; 16.9 c.c. of H_2, and 16.7 c.c. of CO. The 94.8 c.c. of gas originally taken therefore contained

CO_2	O_2	C_2H_4	CH_4	H_2	CO	N_2 (by difference)
4.6 c.c.	.3 c.c.	.2 c.c.	.6 c.c.	16.9 c.c.	16.7 c.c.	55.5 c.c.

The percentage composition by volume therefore is

CO_2	O_2	C_2H_4	CH_4	H_2	CO	N_2
4.8%	.3%	.2%	.7%	17.9%	17.6%	58.5%

The heating value of a fuel gas is best determined directly in a calorimeter (see Chapter XIII). Where a continuous calorimeter, like the Junker, is available, and the gas is under some pressure, it is easy to make arrangements by which the determination is made continuous over the entire time of the test by furnishing facilities for accurately weighing large quantities of water. Where no calorimeter is available, the heating value may be computed by the method outlined and data given in Article 384, in this chapter. In such a case, in order to avoid a complicated and generally unsatisfactory time-average* computation, it is well to see that gas samples are taken for analysis at stated intervals equally spaced over the test, and to make the time of collecting as long as possible. The straight average of all the results may then be taken. The number of samples taken depends largely upon the constancy of the gas composition, fluctuations requiring more frequent analysis.

The ideal method of collecting would, of course, be to arrange an aspirator of the type shown in Fig. 362, p. 515, of sufficient capacity to last for several hours. The sampling may in that way be made practically continuous.

387. Analysis of the Exhaust Gases, and Computation of the Heat Lost in Exhaust. — For methods of collecting exhaust gas samples see Chapter XIII, since the precautions to be observed are the same as for flue gas. Usually the exhaust gases are under some pressure so that no aspirating apparatus is required. As far as the analysis is concerned, what is said in Chapter XIII, with reference to flue gas, applies with equal force to exhaust gases; in fact, if only CO_2, O_2, and CO (N_2 by difference) are tested for, the same apparatus may be used. It has, however, been shown by Güldner and others, that in spite of the fact that there is a considerable excess of air in most gas-engine mixtures, incomplete combustion often results. Güldner cites cases where the heat loss due to combustible components in the exhaust gas amounted to as high as 15

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* A time average is obtained as follows. Assume that on a 12-hour test, samples were taken at the times stated below:

		Interval =
Test begins	8 A.M.	2 hrs.
Sample *A* at	9 A.M	
“ *B* at	11 A.M.	1.5 “
“ *C* at	12 M.	1.5 “
“ *D* at	2 P.M.	2.0 “
“ *E* at	4 P.M.	2.5 “
“ *F* at	7 P.M.	2.5 “
Test ends at	8 P.M.	
		12 hrs.

The analysis of sample *A* is assumed to cover the period from the start of the test to half the interval between samples *A* and *B*. This is 2 hours, from 8 to 10 A.M. Similarly, sample *B* covers the next 1.5 hours, etc. The results of each analysis are then multiplied by this time interval, and the summation of the individual products is divided by the total length of the test, in this case, 12 hours. Since this whole computation is based upon a somewhat unwarranted assumption, its value is doubtful.

per cent of the total heat supplied in the fuel. It must be evident from this that where it is desired to establish a scientifically correct heat balance for a gas-engine test, the exhaust gases should be completely analyzed, and in the field the Heinz or Siebert apparatus may be used for this purpose.

It is stated that the heat loss in exhaust may be found in two ways:

(*a*) By measuring the fuel and the air used, and determining the ratio of fuel to air. With a known fuel composition it then becomes easy to compute the weights or volume of the various products of combustion passing into the exhaust pipe.

(*b*) To analyze the exhaust gases and from the results of this analysis to compute the products of combustion by the method outlined in Chapter XIII for flue gas, or by some other method.

Method (*a*) assumes that there is complete combustion of the fuel, and is correct only under that assumption. How far that assumption may be incorrect has been shown above. In method (*b*) the determination of combustible constituents makes it possible to allow for the effect of incomplete combustion, and on the score of accuracy the latter method is therefore to be preferred to the former. Of course method (*a*), if properly carried out, will give the accurate total weight or volume of the gas passing through the engine under all circumstances, but it may fail to definitely indicate the distribution of this total among the various kinds of gas present, and thereby lead to an erroneous determination of the exhaust loss.

To make the computation of the exhaust loss a general case assume that a complete exhaust gas analysis is made and that some combustible components are found. The method of procedure will then be as follows:

Example. — Fuel used is producer gas with the following composition by volume: 18.73 per cent H_2; 25.07 per cent CO; .31 per cent CH_4; .31 per cent C_2H_4; 48.98 per cent N_2; 6.57 per cent CO_2; .03 per cent O_2. Higher heating value 159 B.t.u. per cubic foot.

The exhaust gases showed by volume:

13.9 per cent CO_2; 5.2 per cent O_2; .9 per cent H_2; 1.4 per cent CO; 78.6 per cent N_2.

Temperature in exhaust = 752° F.

The problem consists of the following parts:

(*a*) The computation of the quantity of the products of combustion.

(*b*) Ratio of air to fuel used, and the excess air.

(*c*) The heat loss in exhaust

For the purpose of comparing them with the results of the exact method for computing the quantity of the products of combustion, as found under (*a*) below, the figures under (*d*) below show the results when the usual approximate method, that is the one assuming complete combustion, is used.

(*a*) *Computation of Quantity of Products of Combustion. Exact Method*

The method used is essentially that developed for the flue gases from furnaces in Chapter XIII, and the student is referred to that chapter for the development of the formulas and equations used

The first step is to convert the per cent by volume composition of the fuel gas to the weight per cent basis. The following table shows the computation involved.

	Composition Per Cent by Volume	Weight per Standard Cu Ft of the Individual Gases	Weight of the Individual Gases in One Standard Cu Ft	Composition, Per Cent by Weight
		Lbs	Lbs	
H_2	18.73	.00561	.00104	1.54
CO	25.07	.07807	.01957	29.02
CH_4	.31	.04464	.00012	.17
C_2H_4	.31	.07809	.00023	.35
N_2	48.98	.07831	.03835	56.89
CO_2	6.57	.12341	.00806	12.00
O_2	.03	.08921	.00002	.03
	100.00		.06730*	100.00

* Summation is weight per standard cubic foot of fuel gas

From the weight analysis, it is now possible to compute the actual weights of carbon and of hydrogen that went through the engine per pound of the fuel gas. *If we next assume that all of the carbon contained in the fuel gas must in some form or other reappear in the exhaust gases, we can at once determine the relation existing between the weight of fuel gas used and the weight of exhaust gases formed by aid of the exhaust gas analysis.*

The only factor that might invalidate this assumption is the carbon deposit in the cylinder. This per pound of gas, however, would be so small as to be negligible.

The actual weight of carbon contained in a pound of the fuel gas is: $\frac{12}{28} \times .2902 = .1244$ pound (from CO content), $\frac{12}{16} \times .0017 = .0013$ pound (from CH_4 content); $\frac{24}{28} \times .0035 = .0030$ pound (from C_2H_4 content); and $\frac{12}{44} \times .12 = .0328$ pound (from CO_2 content); making a total of .1615 pound of carbon (= *c*, see below).

Similarly the actual weight of hydrogen is:

.0154 pound (from H_2 content); $\frac{4}{16} \times .0017 = .0004$ pound (from CH_4 content); and $\frac{4}{28} \times .0035 = .0005$ pound (from the C_2H_4 content): making a total of .0163 pound of hydrogen.

From the equation on p. 535, the relation between the carbon in fuel and that in the flue (exhaust) gas is

$$c = n\left(\frac{44\,CO_2}{3.66} + \frac{28\,CO}{2.33}\right).$$

By substituting in this the value of c as above computed, also the percentage of CO_2 and CO from the flue gas analysis, we will have

$$.1615 = n\left(\frac{44 \times 13.9}{3.66} + \frac{28 \times 1.4}{2.33}\right),$$

from which $n = .00088$.

With this value of n, and in connection with the exhaust gas analysis, we may next compute the actual weight of the various products of combustion by aid of the equations developed on p. 535.

Weight of free O_2 $= .00088 \times 32 \times 5.2 = .1465$ pound.
Weight of CO_2 $= .00088 \times 44 \times 13.9 = .5360$ pound.
Weight of CO $= .00088 \times 28 \times 1.4 = .0345$ pound
Weight of N_2 $= .00088 \times 28 \times 78.6 = 1.9350$ pound.
Weight of H_2 (not burned) $= .00088 \times 2 \times .9 = .0016$ pound.

H_2 actually burned is therefore $= .0163 - .0016 = .0147$ pound, and the water vapor (H_2O) resulting will be $9 \times .0147 = .1312$ pound.

This determines the weight of all of the products of combustion, with the exception of the water vapor as humidity in the air used, which may be neglected.* The sum total is

$$\begin{array}{ccccccccccccc} O_2 & & CO_2 & & CO & & N_2 & & H_2 & & H_2O & & \\ .1465 & + & .5360 & + & .0345 & + & 1.9350 & + & .0016 & + & .1312 & = & 2.78 \text{ pounds.} \end{array}$$

(*b*) *Ratio of Air to Fuel Used, and the Excess Air.*

Since the exhaust gases weigh 2.78 pounds per pound of gas, the air added to each pound of fuel gas to form the mixture must evidently have been 1.78 pounds. The ratio $\frac{\text{air}}{\text{gas}}$ is then $\frac{1.78}{1}$ by weight. The gas weighs .0674 pound, (see table above) and air .0807 pound, per standard cubic foot. This would make the mixture under these conditions consist of $\frac{1}{.0674} = 14.84$ cubic feet of gas.

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* Water brought in as humidity in air amounts to about .001 pound per cubic foot of gas.

and $\frac{1.78}{.0807} = 22.05$ cubic feet of air. This makes the ratio of $\frac{\text{air}}{\text{gas}}$ by volume $= \frac{22.05}{14.24} = \frac{1.49}{1}$.

From equation (12a), p. 859, the theoretical volume of air required by this gas per standard cubic foot

$$= \frac{\frac{.2507 + .1873}{2} + (2 \times .0031) + (3 \times .0031) - .0003}{.21} = 1.12 \text{ cu. ft.}$$

This would indicate an air excess equal to $\frac{1.49 - 1.12}{1.12} = 33$ per cent, by volume. But the result of the computations above has shown that .0345 pound of CO and .0016 pound of H_2 remained unburned per pound of fuel gas, and therefore required no oxygen. These weights amount to .03 cubic foot of CO and .02 cubic foot of H_2 per standard cubic foot of fuel gas. So that only .2207 cubic foot of CO and .1673 cubic foot of H_2 actually burned. This reduces the theoretical air supply necessary to 1.00 cubic foot per cubic foot of gas, and the real excess of air therefore is $\frac{1.49 - 1.00}{1.00} = 49$ per cent, by volume.

(c) *The Heat Loss in Exhaust.* — This is, of course, made up of the heat lost in the combustible components of the gas plus the quantity of sensible heat carried off, measured above some assumed datum temperature, say 32° F.

(1) Heat lost in combustible components:

In CO $= .0345 \times 4{,}380 = 151$ B.t.u.
In H_2 $= .0016 \times 61{,}950 = 99$ B.t.u.

250 B.t.u.

(2) Sensible heat lost.

This is in any case (except that of water vapor) equal to the weight of the particular gas multiplied by the temperature range $= 752 - 32 = 730°$, and by the mean specific heat, C_{pm} between 32° and 752°, as taken from the curve sheet, Fig. 537.

Sensible heat lost in O_2 $= .1465 \times 730 \times .222 = 24$ B.t.u.
Sensible heat lost in CO_2 $= .5360 \times 730 \times .227 = 89$ B.t.u.
Sensible heat lost in N_2 $= 1.9350 \times 730 \times .251 = 354$ B.t.u.

467 B.t.u.

(3) Heat lost in water vapor (see p. 467)

$$= .1312\,(1058.7 + .455\,t_1 - t_2 + 32)$$
$$= .1312\,(1058.7 + .455 \times 752 - 32 + 32) = 184 \text{ B.t.u.}$$

Total heat loss per pound of fuel gas is $250+467+184$ =	901 B.t.u
Higher heating value of gas (computed) per standard cubic foot	159 B.t.u.
Higher heating value of gas (computed) per pound . . .	2360 B.t.u.
Total exhaust gas loss	38.2 per ce
Loss in combustible gases	10.6 per ce
Loss in sensible heat	27.6 per cen

(*d*) *Approximate Method of Computing Quantity of Products of Combustion.* — Where the combustible components of an exhaust gas have not been determined (i.e., are assumed not present), a somewhat shorter method, giving a result which will be more or less correct, depending upon the error involved in such assumption, may be used

In that case, the exhaust gas analysis above given would have been, by volume:

13.9 per cent CO_2; 5.2 per cent O_2, and 80.9 per cent N_2 (by difference).

From equation (15a), p. 859, the total number of cubic feet of N_2 that would have resulted from the combustion of one cubic foot of the fuel gas *with the theoretical air supply of* 1.12 *cubic feet per cubic foot of gas*, is

$$N_2 = (.79 \times 1.12) + .4898 = 1.3746 \text{ cu. ft.}$$

Of this total, $\frac{.8848}{1.3746} = 64.37$ per cent is due to air used; $\frac{.4898}{1.3746} = 35.63$ per cent is due to N_2 already in fuel gas.

The computation for total air really supplied may now be made as follows:

Total N_2 from exhaust gas analysis $= 80.90$

Of this amount, N_2 due to excess air (as measured by free O_2)

$$= \frac{79}{21} \times 5.2 \qquad = 19.60$$

Leaves N_2 due to fuel gas and to air actually needed 61.30

Of this remainder, 35.63 per cent is due to N_2 content in fuel gas $= 21.85$

Leaves N_2 due to air actually needed 39.45

From this data

$$\text{Excess coefficient} = \frac{N_2 \text{ due to excess air} + N_2 \text{ in air needed}}{N_2 \text{ in air needed}},$$

$$= \frac{19.60 + 39.45}{39.45} = \frac{59.05}{39.45} = 1.50.$$

This result would show that the volume of air supplied per cubic foot of gas is $1.50 \times 1.12 = 1.68$ cubic feet, and that the volume ratio is $\frac{1.68}{1}$ instead of $\frac{1.49}{1}$, as the exact method shows. The discrepancy is primarily due to the fact nd ne approximate method is compelled to assume that the air actually used ıe combustion is that shown to be the theoretical volume necessary, i e., cubic feet. The fact is, however, that on account of incomplete combusı, the real volume of air used up is only 1.00 cubic foot per cubic foot of fuel s. Therefore this is the figure which should have been multiplied by the cess coefficient 1.50 to obtain the total air supply, ıd we have 1.50 × 1.00 = 1.50 cubic feet, which grees very closely with the exact result.

The computation by the approximate method here primarily made to indicate what the degree error involved in some cases may be. Of ›urse not having the complete analysis of the haust gases, we are compelled to assume comete combustion, i.e., that the theoretical amount air that was required is 1.12 cubic feet (in this se). The result is that the ratio of $\frac{\text{air}}{\text{gas}}$ is dermined too large by $\frac{1.68 - 1.49}{1.49} = 12.8$ per cent.

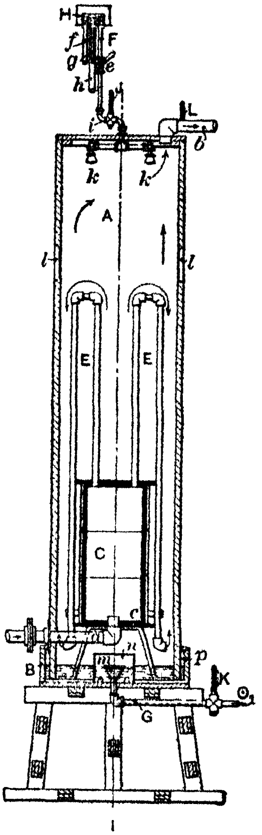

Fig. 547.—Staus Exhaust Gas Calorimeter.

388. Experimental Determination of Exaust Loss. — The heat lost in exhaust has so been experimentally determined. taus* constructed an exhaust gas calometer of which the details are shown in ig. 547. The gases find their way from ıe exhaust pipe *a* into the muffler *C*. he latter is enclosed in the calorimeter, hich consists essentially of a wooden box lined with galvanized iron. The lower ıd of the box is open and rests in the ough *B*, in which a water-seal of constant ·ight is maintained by means of the ·erflow funnel *m*. The exhaust gases pass from *C* into six pes *E*, arranged as shown, and escape into the box *A* through bows at the lower end. The box being sealed below, the funnel ›ening also being covered by the hood *n*, the gases rise in *A* and ıally escape through *b*. In their ascent they meet fine streams of ıter produced by the spray nozzles *k*, the water-supply head being maintained constant by the constant head apparatus *F*. Valve *e* controls the rate of supply. Thermometers at *J* and *K* determine the temperature of water entering and leaving; thermometer *L*, the sensible heat still remaining in the gas above any assumed datum. Two small windows *l* are furnished to watch the operation of the spray nozzles. Means must, of course, be furnished to accurately determine the quantity of water used.*

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* A. Staus, *Zeitschrift des Vereins deutscher Ingenieure*, May 3, 1902.

The method of computation is obvious. It should be noted that the results obtained can be accurate only as long as the gases contain no combustible parts. If these are present, the heat of combustion in them will escape through *b* unaccounted for.

389. The Determination of the Quantity of Producer Gas Made per Pound of Fuel. — There are two methods open, one of which may be called mechanical, the other chemical.

The mechanical method consists in the use of gas meters, Venturi meters, Pitot tubes, etc. Concerning these instruments the student is referred to Chapter XII, for discussions regarding theory, accuracy, precautions necessary, etc. It may be said that where the flow of gas from a producer is fairly steady, i.e., in a pressure plant, the means above mentioned will give excellent service if properly used, but in a suction-gas plant, the difficulties encountered on account of the rapid fluctuations of velocity, are in most cases insurmountable. In pressure-gas plants the gasometer method, which consists in determining the rate of fall of the gasometer bell while the gas inlet is closed (purge pipe on producer open) and the engines or other apparatus are drawing on the gasometer, is sometimes used. Unless the demand is fairly constant, these determinations have to be made often in order to obtain an average rate of fall. This coupled with the difficulty of accurately rating the capacity of the gas holder for a given drop, especially in large sizes, operates against the accuracy of this method, although it is often the only one available.

The *chemical method* of determining the volume of gas made is of interest and of general applicability. It depends upon the fact that all of the carbon contained in a fuel used in a gas producer must reappear in some form or other in the following four ways: (*a*) as gaseous carbon in CO and in CO_2 in the producer gas, or in CH_4, C_2H_4, etc., as distillation products in the gas; (*b*) as solid carbon in the refuse (ash) from the producer; (*c*) as gaseous carbon in the hydrocarbon gases (tar) not included in the distillation products under (*a*); and (*d*) as dust or soot carried mechanically out of the producer by the gases and deposited in the pipes. All of these items are susceptible of determination with a degree of accuracy sufficient for engineering work.

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* An exhaust gas calorimeter of similar construction is now on the market, made by Junkers & Co., Dessau.

If we let C_f represent the weight of total carbon in a pound of the fuel used, as determined by chemical analysis;

C_g the total weight of carbon that can be accounted for, by computation from chemical analysis, in a standard cubic foot of the producer gas made;

C_r the weight of carbon which appears in the refuse per pound of fuel fired;

C_t the weight of carbon that appears in tar per standard cubic foot of producer gas;

C_s the weight of carbon appearing in soot or dust per standard cubic foot of producer gas; and

V the number of standard cubic feet of producer gas made per pound of fuel fired,

we must have the relation

$$C_f - C_r = V(C_g + C_t + C_s),$$

from which

$$V = \frac{C_f - C_r}{C_g + C_t + C_s} \text{ standard cu ft. per lb. of fuel fired.} \qquad (31)$$

C_f and C_r are determined from the chemical analysis of fuel and refuse. See Chap. XIII. C_g is computed from the volumetric chemical analysis of the producer gas. The determination of C_t and of C_s needs further consideration.

Determination of Tar and Soot. — There are several types of apparatus which may be used for the quantitative determination of tar. In some of them the tar is taken out of the gas by means of alcohol, in others some kind of a dry filter (dried asbestos fiber or wool) is used. The method given by Tieftrunk* belongs to the first class.

The apparatus is shown in Fig. 548. It consists of a cylinder *a* which is fitted with a tight-fitting cover *b* held on by clamps ***kk***. The tube *cg*, through which the gas enters, reaches nearly to the bottom of *a*. Over the lower end of *g* there are slipped five or six bell-shaped brass plates *h*, perforated with holes 1.5 mm. in diameter and 5 mm. apart. Cylinder *a* is filled with alcohol (30 to 35 per cent by volume) to a height sufficient to cover the last plate. The gas, after bubbling up through the alcohol, leaves through *d* and passes next through a U-tube *f* filled with cotton. The last cylinder in the series, which the gas enters at *e*, is filled at *o* with cellulose, and upon this rests a column *mm* of bog-iron ore, separated from the cellulose by a filter paper. This cylinder is used to take H_2S out of the gas, which is done for the purpose of protecting the delicate gas meter connected to the outlet *p*. The gas is drawn through the entire apparatus by means of some type of aspirator pump connected beyond the gas meter. In case of producer gas, the use of the H_2S absorption apparatus may be dispensed with, and the same may be done for any gas if some device other than a gas meter, as for instance a gasometer, be used to draw and measure the gas.

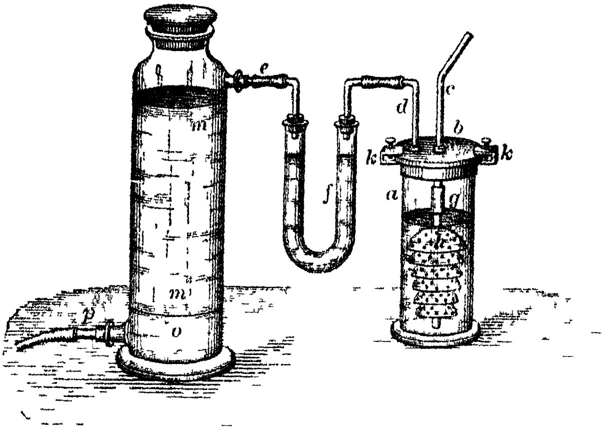

FIG. 548.—TIEFTRUNK APPARATUS FOR TAR DETERMINATION

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* Winkler, Industriegase, Vol. II, p. 51; also Hempel-Dennis, Gas Analysis.

Since the gas sample for this test must be taken close to the producer, ahead of the scrubber, the gas is apt to be too hot to be taken directly into the apparatus, and for that reason a glass condenser (not shown in Fig. 548, see *E* in Fig. 549) is interposed between the tube *c* and the gas-sampling tube in the gas flue. The porcelain sampling tube is open at the end, and is introduced into the gas main through a gas-tight stuffing box. The end of the sampling tube must, of course, face the gas stream. The connection between sampling tube and condenser cannot be made by rubber tubing, on account of the heat, and it is usually necessary to make some kind of a coupling with a stuffing gland at each end. The condenser is tilted toward the alcohol cylinder, so that the tarry gases condensing in it will flow into the cylinder by gravity as far as possible.

The minimum quantity of gas that should be passed through the apparatus is about 20 cubic feet, the suction pump being so regulated that not more than 1.5 to 2.0 cubic feet passes per hour. The soot and tar collect in the sampling tube, the condenser, and in the alcohol cylinder. If any of the tar should pass the latter, the cotton in *f* will be colored brown. At the end of the test, after noting the exact volume of gas that has passed, together with pressure and temperature at the meter, the apparatus is taken apart and transported to the chemical laboratory. Here all traces of soot and tar in condenser and sampling tube are carefully washed out with alcohol and added to the contents of the cylinder (the tar is extracted from the cotton by means of carbon disulphide, if that should be necessary). The chemical determination of the amount of carbon carried by the material collected is then made. Since neither the operation of washing, nor extraction, can be done by the engineer in the field, and since the determination of the carbon must be done by a chemist, no further directions will be given, but the student is referred to books on chemistry.

The second type of apparatus is simpler, and by means of it, tar, soot, and water vapor may be determined. It is known as Lord's apparatus, Fig. 549. The following description of its action is taken from Wyer.*

* S. S. Wyer, "Producer Gas and Gas Producers."

" B is the sampling tube made of 0.5-inch pipe which is placed in the gas flue; A is an annular jacket surrounding it, and has pipe connections at D and C. Live steam is blown in at D, and out at C, the object of this being to keep the temperature of the iron pipe below the point at which the iron would act on CO_2. This will secure a sufficient cooling, and yet will leave the temperature high enough to prevent the condensation of moisture. E is an ordinary condenser through which cold water is circulated. F is a small flask filled with ignited asbestos fiber and containing a thermometer G. J and L are tanks filled with water and connected at K. I is a valve. H is a rubber tube, connecting J and F. Q is a thermometer placed in a stopper in a pipe with valve R, the object of this valve being to make it possible to remove the thermometer when gas is in the tank J. M is a float to which is fastened the curved tube N, which acts as a siphon and which has a small nozzle O, with a pinchcock P on the rubber connection. The object of the float and tube is to keep a constant head above the nozzle, and thus insure a uniform flow through it. The operation of the tank is as follows: Disconnect the rubber tube H and fill the tanks J and L with water until they overflow at the valve I; fill the siphon N with water and close the stopcock P, attach the rubber tube H to stopcock I, circulate water through the condenser E, and steam through the water jacket A. Then open valve P; the water will be drawn out of tanks L and J, and the gas will be drawn through

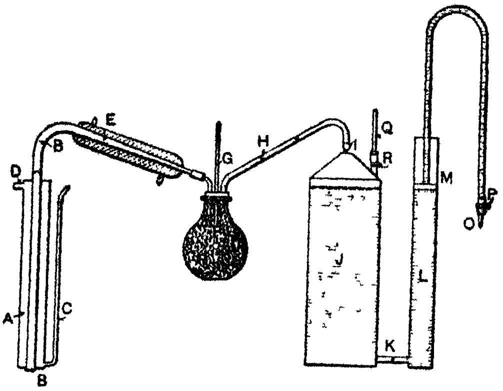

FIG. 549.—LORD'S APPARATUS FOR TAR AND SOOT DETERMINATION.

condenser *E*, flask *F*, and tube *H* into the top of the tank *J*. The water in excess of the saturation of the gas at the temperature of the small flask is condensed, and any tar and soot in the gas retained in the ignited asbestos in the flask. After the test, the flask and its contents are weighed, and the increase over the weight taken before the test gives the quantity of the tar and water condensed from the volume of the gas which has passed through the flask. This volume is determined by measuring the quantity of water which had run out of the aspirating tank *J*.

"The quantity of water remaining in the gas, after passing out of the little flask used as a receiver, is calculated from the temperature of the issuing gas, which was saturated with water vapor. The water in the gas is then the sum of the permanent vapor and that condensed. The water in the flask is determined by drying the contents over sulphuric acid to constant weight and determining the loss. The dry contents are then ignited and the further loss of weight estimated as soot and tar.

"Let B = barometric pressure.
Tt = temperature of gas in tank.
Tb = temperature of gas in flask.
Vt = volume of wet gas in tank at temperature Tt.
Vs = Vt reduced to 32° F. and 14.7 pounds per square inch.
Vd = volume of dry gas at 32° F. and 14.7 pounds per square inch.
Bt = aqueous tension of water vapor corresponding to Tt.
Bb = aqueous tension of water vapor corresponding to Tb.
W = weight of 1 cubic foot of water vapor corresponding to Tb.
Wb = weight of water vapor condensed in flask.
Wt = weight of permanent water vapor in volume Vs.

$\frac{Bb}{B}$ = percentage by volume of water vapor in flask.

$\frac{Bt}{B}$ = percentage by volume of water vapor in Vs.

$$Vd = Vs\left(1 - \frac{Bt}{B}\right).$$

$Vs\frac{Bb}{B}$ = total volume of permanent water vapor in Vs.

$$Vs\frac{Bb}{B}W = Wt.$$

$Wt + Wb$ = total weight of water carried in volume, Vd, of gas."

If in this apparatus a porcelain collecting tube instead of the iron be used, the heater may be dispensed with, in case the test is made on raw (hot) producer gas. Further, for the purpose in view, which is the determination of the weights of carbon in the tar and soot, this procedure must be changed in so far that, after drying the flask to constant weight, the percentage of C in the excess weight remaining must be determined by chemical means. It need hardly be said that, where a good meter (of the type of the Junker calorimeter) and a suction pump is available, vessels *J* and *L* may be dispensed with.

The following is an example of the chemical method of determining the volume of gas made per pound of fuel.

Analysis of coal: H_2O, 1.61 per cent; C_2, 74.95 per cent; H_2, 4.41 per cent; S, 1.17 per cent; ash, 14.09 per cent.

Analysis of gas: CO_2, 2.11 per cent, by vol.; C_2H_4, .30 per cent; O_2, 2.63 per cent; CO, 25.83 per cent; CH_4, 2.10 per cent; H_2, 8.10 per cent; N_2, 58.93 per cent.

Analysis of ash: 10 per cent C_2.

Percentage of refuse is 15.60.

Tar and soot determination: 15.96 cubic feet of gas (at a vacuum of 3 inches of water and a temperature of 80° F.) showed .3909 gram of C in tar and soot.

Here C_f = .7495 (see page 885).

C_g is found as follows (the gases carrying carbon being CO_2, CO, CH_4, and C_2H_4):

Kind of Gas.	Cubic Feet Present in 1 Standard Cubic Foot of Producer Gas	Weight of Carbon in 1 Standard Cubic Foot of the Kind of Gas	Weight of Carbon in Each Kind of Gas Present.
CO_2	.0211	.0334 *	.00070
CO	.2583	0334	.00868
CH_4	0210	0334	.00070
C_2H_4	0030	.0668	00020
			C_g = 01028

* To show the derivation of this, we have 1 lb C produces 3.66 lbs CO_2; and under standard conditions the weight of CO_2 = .12341 lb per cu ft Hence the weight of carbon per cu. ft = $\frac{1}{3.66 \times 12341}$ = 0334 lb.

C_r, the weight of carbon lost in ash = 10 per cent of .1560 = .0156 pound per pound of fuel.

$$C_t + C_s = \frac{.3909}{453.6 \times 15.96} = .00005 \text{ pound per cubic foot of gas.}$$

Here 453.6 is the number of grams per pound. The volume of gas (15.96 cubic feet) has not been changed to standard conditions in this case, as the correction would not amount to anything, in view of the small weight of carbon.

The gas made per pound of coal from the above data and equation (31) will then be

$$V = \frac{C_f - C_r}{C_g + C_t + C_s} = \frac{.7495 - .0156}{.01028 + .00005} = 71.0 \text{ standard cubic feet.}$$

390. The Testing of Gas Producers and of Gas Engines. — A code for testing gas engines was adopted and published by the American Society of Mechanical Engineers as an Appendix to the "Steam Engine Code" in Vol. XXIV (1902) of the Transactions. This was followed in 1905 by a code adopted by the British Institution of Civil Engineers. Finally in 1906 the Verein deutscher Ingenieure adopted a code which was translated by Mr. F. E. Junge and published in *Power*, Feb., 1907. The latter code also takes up the matter of testing gas producers.

While these codes contain a lot of valuable information, it is true, particularly of the American code, that the information given in many cases is not sufficiently detailed to guide a student completely. These matters have therefore been taken up at greater length in the preceding paragraphs in this chapter, and their discussion will be completed in the paragraphs following. Other provisions of the code (those referring to the calibration of instruments, determination of brake horse-power, etc.) have already been fully discussed in other parts of this book. It is therefore judged best not to publish any of the gas-engine codes in full, but to set forth the main viewpoints determining the arrangements necessary for a test of a gas producer or a gas engine, and to incorporate into the discussion those parts of the codes that are pertinent and are not covered in any other parts of this book.

391. Directions for Testing a Gas Producer. — The *object of the test* is, in general, to determine either efficiency, or capacity, or both. Secondary objects may be to find the most suitable fuel for any given type of producer, to determine the best ratio of water to air for any given fuel, to determine the tar and water-vapor content of the gas made, etc.

The capacity of a producer is usually stated either as the horse-power it is capable of supplying in gas engines, or as the number of pounds of coal that each square foot of producer-grate area will convert into gas per hour (so-called gasification capacity). Since engines differ as to the amount of gas required per horse-power hour, and different fuels show quite different gasification qualities, it must be evident that either method of rating capacity requires definite specifications as to engines or fuels. This is a matter that must be carefully covered in the guarantee.

Capacity tests are a simple matter as it is generally only necessary to load up a producer either by sufficient engine capacity or by removing the gas made in some other manner and observing whether the plant will stand up to this service for the time specified in the contract. Measurement of engine output, or of fuel consumed, is really all that is required.

It will be assumed, however, that it is desired to make an efficiency test and to establish, as far as possible, a complete heat balance. For that purpose the following requirements should be met and arrangements must be made to determine the following items. In any given case, modifications and simplifications may be introduced to suit the particular object in view.

1. *Examine the producer* for defects and remedy these if a maximum efficiency test is to be made. Determine the dimensions of the grate, the contour of the walls, and make sketches of any unusual constructive details.

2. *Duration of Trial.* — The viewpoints which control this are very much the same as for a boiler test. On account of the difficulty of properly judging the contents of a producer, a trial should never be less than 8 hours in duration, and preferably not less than 12. It depends, of course, upon the usage in the particular plant if a plant test is desired, as to whether a test can be extended beyond 8 or 10 hours or not.

3. *Starting and Stopping.* — The producer should be thoroughly heated before starting a test, i. e., it must be in operation for a sufficient length of time(not less than 24 hours)under normal conditions, and if possible on the same grade of fuel, to allow of the attainment of average normal temperature in brick work, flues, etc. The condition of the fuel bed must, of course, be the same at the end as at the beginning. Note, at the beginning, the position of the ash

and incandescent zone, the upper level of the fuel bed, and its condition with respect to covering of green fuel, etc. During the trial, maintain these conditions as nearly as possible, cleaning out ash, poking fire, etc., the same as in ordinary operation, and at the end have the conditions the same as at the beginning, as nearly as possible. Clean out the ash pit immediately before the beginning and before the end of the trial. If it is not possible to remove ashes and clinker during the trial, shut down the producer immediately after the expiration of the time, clean out the refuse, and bring the fuel bed to the initial condition at the start. The fuel so used must be added to that used during the trial.

If the plant uses an auxiliary boiler to furnish steam to the producer, the start and stop of the test with respect to this auxiliary should be managed exactly as laid down in the code for boiler testing.

4. *Conducting the Trial.* — Arrangements must be made to make the demand for gas uniform, as far as possible. Uniformity of conditions should also prevail with respect to steam pressure and air blast (in the case of a pressure plant); quantity of water supplied to the air (in case of a suction plant); thickness of fuel bed and ash column; frequency of firing and amount of fuel fired; poking fire and interval between cleaning fires.

Keep as complete a record as possible. Record the time of every observation of weight, temperature, etc., and take note of every event, however unimportant it may seem at the time.

5. *Observations and Calculations Required.* — (*a*) Weight of fuel and refuse. The sampling is carried out as outlined under Boiler Tests.

In some types of producers the raking out of ash may bring down some unburned fuel. This may either be returned to the producer or its weight may be subtracted from the weight of fuel used. Any unburned fuel, however, that falls through the grate while the producer is operating normally must be counted as refuse. Neither can an allowance be made for any fuel dust that gathers in the pipes and flues during a long-continued test.

Any fuel used by an auxiliary boiler for furnishing steam must be added to the fuel used by the producer.

Laboratory samples of fuel and refuse must be taken in the usual way. In some cases the ash is wet down during drawing, or it is wet as it leaves a producer of the water-bottom type. In such cases, determine the weight of the wet ash, after allowing the water to drain off, and then immediately take a laboratory sample in order to be able to allow for the water content and thus to obtain weight of normal ash.

(*b*) Weight of water supplied to the producer. In case of a pressure plant, this is found by weighing the feed water to the auxiliary boiler and determining the quality of steam at the producer. When the steam for blowing purposes is taken from a battery of boilers which are also used for other purposes, nozzle or orifice measurement of the flow in the supply pipe to the producer may be resorted to.

In a suction-gas plant the water supplied the vaporizer must be accounted for. If the vaporizer is fitted with an overflow to maintain constant level, the water wasted, together with its temperature, must be determined.

Another way to determine the steam or water supplied to a producer of either type is based upon the consideration that the hydrogen and water vapor in the resulting gas must come from the following sources: moisture in coal, hydrogen gas in coal, humidity in air used, and steam supplied. The supply from all of the sources but the last can be easily found, so that the last may be determined by difference.

(*c*) Weight or volume of air supplied. This measurement is difficult, in some cases impossible, to make directly. The weight of the air that must have been supplied can usually be obtained by computation on the basis of gas or fuel analysis and volume or weight of gas produced. This computation is similar to that made for flue gas. See Chapter XIII.

(*d*) Humidity of air supply. This should be determined in every case. It may be found by the wet and dry bulb thermometer or some other form of hygrometer. See Chapter XXIII.

(*e*) Analysis of fuel and gas, and heating value of gas. A complete producer test requires the ultimate analysis of the fuel used. See Chapter XIII. The methods of sampling and analyzing the gas have already been discussed in Art. 386. The gas should be examined for tar and dust. See Art. 389.

The heating value of the gas should, if possible, be determined during the test by means of a calorimeter.

(*f*) Volume or weight of gas made. This may in some cases be directly measured, but can in all cases be computed with fair accuracy on the basis of fuel, gas, ash, and tar analysis. See Art. 389.

(*g*) Temperature observations. The following temperatures should be taken: of external air and of air in the producer room, air entering and leaving the economizer or preheater, if one is used; of mixture of air and steam entering the producer; of gas leaving the producer, leaving the economizer, leaving the washing apparatus (or in the gas main); of feed water to auxiliary boiler in a pressure plant; or of water entering vaporizer in a suction plant (and leaving overflow from vaporizer if necessary).

All of these temperatures can usually be obtained with the ordinary mercury thermometer, except the temperature of the gas leaving the producer, which will require a pyrometer.

(*h*) Pressure observations. These should be as follows: barometer in producer room, steam pressure (in pressure plant), pressure in inches of water or mercury of mixture of air and steam entering producer, of gas leaving producer, and of gas in main beyond the washing apparatus.

The following form for recording observations and the results of computations based upon them is, with a few minor changes, given by S. S. Wyer in his book on " Producer Gas and Gas Producers."

DATA AND RESULTS OF GAS PRODUCER TEST.

General Data.

1. Test made by. ....
2. Test made to determine..
3. Chemical analyses made by ....
4. Type of producer..
5. Producer built by..
6. Date of installation..
7. Kind of fuel ....
8. Form of grate..

General Data.—Continued.

9. Form of ash pit..........
10. Form of blower..........
11. State of weather..........
12. Barometer in producer room..........
13. Date of test..........
14. Duration of test..........

Dimensions of Producer.

A complete description and drawings of producer should be given on an annexed sheet.

15. Grate surface ....Width..........Length..........
Diameter..........Area..........
16. Height of bed of ashes..........
17. Height of top of fire above grate..........
18. Thickness of ash zone..........
19. Position of air pipes..........
20. Position of steam inlets..........
21. Diameter of producer..........
22. Inclination of bosh wall..........

Average Pressures.

23. Steam pressure near nozzle (lbs. per sq. in.)..........
24. Force of draft in ash pit (in. of water)..........
25. Force of draft in gas flue (in. of water)..........
26. Steam pressure in auxiliary boiler (lbs. per sq. in.)..........

Average Temperatures

27. Of external air..........
28. Of producer room..........
29. Of steam near nozzle..........
30. Of air entering preheater..........
31. Of air entering producer..........
32. Number of degrees of preheating..........
33. Of escaping gases from producer..........
34. Of escaping gases from economizer..........
35. Of ash pit..........
36. Of feed water entering auxiliary boiler..........
37. Of water entering vaporizer..........
38. Of water leaving vaporizer..........

Fuel.

39. Size and condition
40. Total weight of fuel fired
41. Percentage of moisture in fuel
42. Total weight of dry fuel consumed
43. Total weight of ashes as drawn out
44. Percentage of moisture in ashes
45. Total weight of dry ashes
46. Percentage of carbon in dry ashes
47. Total combustible consumed
48. Percentage of incombustible in dry fuel
49. Total combustible consumed in auxiliary boiler
50. Total combustible required to generate the steam used in producer, if the producer was used without its own auxiliary boiler
51. Total amount of combustible used in the production of the gas

Proximate Analysis of Fuel.

	Of Fuel, Per Cent		Of Combustible, Per Cent.
52. Fixed carbon			
53. Volatile matter			
54. Moisture			
55. Ash			
	100 per cent		100 per cent

56. Sulphur, separately determined

Ultimate Analysis of Fuel.

	Of Fuel, Per Cent		Of Combustible, Per Cent.
57. Carbon (C)			
58. Hydrogen (H)			
59. Oxygen (O)			
60. Nitrogen (N)			
61. Sulphur (S)			
62. Ash			
	100 per cent		100 per cent

Moisture in sample of fuel as received.

Analysis of Ash and Refuse.

63. Carbon..
64. Earthy matter...

Consumption of Fuel.

65. Total fuel consumed per hour in running producer......................
66. Total combustible consumed per hour in running producer..............
67. Dry fuel per square foot of grate surface per hour consumed in producer itself...

Calorific Value of Fuel.

68. Calorific value by oxygen calorimeter per pound of dry fuel......B.t.u.
69. Calorific value by oxygen calorimeter per pound of combustible....B.t.u.
70. Calorific value by analysis per pound of dry fuel ....B.t.u.
71. Calorific value by analysis per pound of combustible............B.t.u.

Quality of Steam.

72. Percentage of moisture in steam..................................
73. Number of degrees of superheating...............................

Quantity of Steam.

74. Actual weight of steam per hour..................................
75. Ratio of steam to air supply......................................

Quantity of Air.

76. Absolute humidity...
77. Actual weight of air per hour.....................................

Water.

78. Total weight of water used in vaporizer...........................
79. Number of heat units carried out per pound of fuel..................

Efficiency.

80. Grate efficiency of producer.....................................
81. Hot-gas efficiency...
82. Cold-gas efficiency..

Cost of Gasification.

83. Cost of fuel per ton delivered in producer room......................
84. Cost per British thermal unit in gas...............................

Poking.

85. Method of poking ....
86. Frequency of poking...

Firing.

87. Method of firing..
88. Average intervals between firing..
89. Average amount of fuel charged each time................................

Gas Analysis.

Per Cent.

90. Carbon dioxide (CO_2)...
91. Carbon monoxide (CO)..
92. Oxygen (O)..
93. Hydrogen (H)..
94. Marsh gas (CH_4)..
95. Olefiant gas (C_2H_4)...
96. Sulphur dioxide (SO_2)..
97. Nitrogen (N) by difference..

100 per cent

98. Pounds moisture in gas per pound of fuel..............................
99. Pounds soot and tar in gas per pound of fuel..........................
100. Calorific value of gas from analysis.................................
101. Calorific value of gas determined with calorimeter...................
102. Specific heat of gas...
103. Figure of merit of gas...
104. Carbon ratio $\frac{C}{H}$...
105. Volume of gas per pound of fuel......................................

In this table the method of obtaining all of the items with the possible exception of the efficiencies (items 80 to 82) and of the figure of merit (item 103) should be clear. These will be considered in the next article in connection with the heat balance.

392. The Heat Balance of a Gas Producer. — There are two methods of establishing a heat balance for a gas producer. In the first, the heat interchanges in the endothermic and exothermic reactions occurring in the process are computed and a balance established between these in connection with the heat supplied in air and steam and that accounted for in sensible heat of the gas. The item necessary to complete the balance is the radiation loss. This method is scientifically very interesting but rather involved.*

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* See an article by K. Wendt on "Untersuchungen an Gaserzeugern," in the *Zeitschrift des Vereins deutscher Ingenieure*, Nov. 28, 1904. Detailed computations are there given.

The second method is much simpler and serves very well for all ordinary purposes. It is based simply upon the principle of (I) accounting for all the heat supplied to a producer, and (II) accounting for all the heat that reappears in gas, etc. The discrepancy between these two accounts is assumed to be the loss by radiation, conduction, etc. The heat balance is nearly always established on the basis of unit weight of fuel.

(I) *Heat Supplied to Producer.* — This consists of the heat in the coal and of the heat in the air-steam mixture per unit weight of coal.

(1) Heat value of unit weight of fuel $= Q_1$ B.t.u. (32)

(2) Heat in air-steam mixture consists of three parts: (*a*) heat in air; (*b*) heat in humidity in air, and (*c*) heat in steam supplied. All these are computed above some reference temperature t_r, which is usually taken at the temperature of the room that obtained during the test. The mixture has a final temperature, t_m, as it enters the producer, while the pressure is either slightly below or slightly above barometer pressure, depending upon the type of producer. Under item (*c*) the weight of steam is determined as outlined on page 894. It may be assumed that this steam is superheated as it enters the producer. The computations are as follows:

(*a*) Heat in air supplied

$$= G_a C_{pm} \, (t_m - t_r) = Q_2 \text{ B.t.u.}, \tag{33}$$

where G_a = weight of dry air per unit weight of fuel supplied to producer alone, and

C_{pm} = mean specific heat between t_r and t_m.

(*b*) Heat in humidity in air

$$= G_h C_{pm} \, (t_m - t_r) = Q_3 \text{ B.t.u.}, \tag{34}$$

where G_h = weight of water vapor as humidity carried by G_a pounds of air, and

C_{pm} = mean specific heat of the water-vapor, which for all practical purposes may here be assumed = .46.

(*c*) Heat in steam supplied

$$= G_s \{ \lambda + C_{pm} \, (t_m - t_s) - (t_r - 32) \} = Q_4 \text{ B.t.u.}, \tag{35}$$

in which G_s = weight of steam supplied per unit weight of fuel,
λ = total heat in steam at the *partial pressure* due to it in the blast or suction pipe,
C_{pm} = mean specific heat between t_m and t_s, in which the latter is the saturation temperature at the same pressure for which λ was found.

For all practical purposes, the expression

$$G_s(1090.7 + .455\,t_m - t_r) = Q_4 \text{ B.t.u.} \qquad (35a)$$

may be substituted for the above expression. See Chap. XIII.

(II) *The Heat Accounted for.*—This appears in the following items: (1) heat value of the clean gas; (2) total heat of the water vapor in the gas as made; (3) heat of combustion of the tar in gas as made; (4) heat of combustion of the soot in gas as made; (5) sensible heat of the gas and its impurities; (6) heat value of the fuel in refuse; (7) sensible heat of the refuse, and (8) heat lost in radiation, conduction, etc.

(1) Heat value of the clean gas

$$= VH = Q_5 \text{ B.t.u.}, \qquad (36)$$

where V = the number of standard cubic feet of gas produced per pound of fuel supplied to producer, and
H = the heat value in B.t.u. per standard cubic foot.

(2) Total heat of water vapor in gas

$$= G_{s'}\{\lambda + C_{pm}(t_g - t_{s'}) - (t_r - 32)\} = Q_6 \text{ B.t.u.}, \qquad (37)$$

in which $G_{s'}$ = weight of water vapor carried by V cubic feet of gas,
λ = the total heat in steam at the partial pressure due it as the gas leaves the producer,
t_g = temperature of gas leaving the producer,
C_{pm} = mean specific heat between t_g and $t_{s'}$, the latter being the saturation temperature for which λ was determined.

Here again, as for (I, *c*) above, the expression

$$G_{s'}(1090.7 + .455\,t_g - t_r) = Q_6 \text{ B.t.u.} \qquad (37a)$$

may be substituted with sufficient accuracy.

(3) and (4). Loss in tar and soot. These are usually determined together. The loss is somewhat indeterminate, because the heat value of the tar varies with its composition. It may be expressed by

$$G_tH_t = Q_7 \text{ B.t.u.,} \tag{38}$$

in which G_t = the total weight of tar and soot determined in V cubic feet of gas, and

H_t = heat value of tar and soot, which may be assumed at about 14,000 B.t.u. per pound.

(5) Heat loss in sensible heat in gas and impurities. This loss is computed from the weight of the individual gases (CO, H_2, CO_2, N_2, CH_4, C_2H_4, etc.) made per pound of fuel supplied the producer, and for each one of these is equal to

$$G_gC_{pm}(t_g - t_r) = Q_8 \text{ B.t.u.,} \tag{39}$$

where G_g = weight of each gas per pound of fuel supplied to producer,

C_{pm} = mean specific heat between gas temperature t_g and room temperature t_r.

Another method is to compute the average C_{pm} for unit weight of the producer gas, and to substitute this in equation (34), in which case G_g = total weight of gas made per pound of fuel. It should be noted that under this head the sensible heat of the water vapor, if the gas should contain any, is not considered, having been taken care of under (II, 2), equation (37) or (37a). The sensible heat lost in tar and soot is a negligible quantity in most cases.

(6) Heat value of the fuel in refuse. This computation is made in exactly the same way as for the similar loss in boilers. See Chap. XVII. On the assumption that the combustible part of the refuse is coke, the loss may be put equal to

$$14{,}540(A - B) = Q_9 \text{ B.t.u.,} \tag{40}$$

in which A = the weight of refuse (dry) per pound of coal, and

B = the weight of true ash (as found by calorimeter) per pound of coal.

(7) Sensible heat in refuse. This quantity is difficult to deter-

mine on account of uncertainty as to the specific heat of the ash and of the temperature of the ash as it leaves the producer. If the ash is allowed to rest in the ash pan for a considerable period, a great deal of the heat originally carried down by it is likely to be returned to the producer by upward currents of air. In any case the loss through this source is small and may be neglected.

(8) Heat lost by radiation, conduction, etc., is determined as the difference between the heat supplied and the heat accounted for, and may be expressed by

$$R = \Sigma Q_{1 \text{ to } 4} - \Sigma Q_{5 \text{ to } 9}, \tag{41}$$

393. **Computation of Gas Producer Efficiency.** — The commercial efficiency of a producer varies not only with the manner in which the various actions and reactions in the process are carried out, but also with the use to which the gas is to be put. The gas as it leaves the producer carries two very distinct quantities of heat, that bound as chemical energy in the combustible part of the gas, and that designated as sensible heat, due to its temperature above some reference temperature. There are commercial uses, as for instance the heating of steel furnaces, reheating furnaces, etc., in which we should strive to utilize, as fully as possible, both of these quantities of heat. There are others, notably the operation of gas engines, in which we must have a cold gas in order to realize full engine capacity. Incidentally, we get this cold gas because the gas must be thoroughly washed for engine operation. In such a case the sensible heat is evidently almost entirely lost. The part not used in a preheater is carried away by the scrubber water. In the first case the *useful effect* is the sum of the heat of combustion plus the sensible heat; in the last it is practically only the heat of combustion. Evidently, since *efficiency* is the quotient of useful effect divided by heat supplied to produce this effect, the efficiency will be different in the two cases. That efficiency which credits the producer with the sensible heat in the gas is known as the *hot-gas efficiency*, that which does not is called the *cold-gas efficiency*.

Let H_1 = heat value of one pound of the fuel as fired,
H_2 = total quantity of heat of combustion in the gas made per pound of fuel, and

H_3 = the sensible heat above some reference temperature (usually room temperature) in the gas made per pound of fuel as it leaves the producer.

Then

$$\text{Cold-gas efficiency} = \frac{H_2}{H_1}, \tag{42}$$

$$\text{Hot-gas efficiency} = \frac{H_2 + H_3}{H_1}. \tag{43}$$

Referring to the previous article, H_1, in a suction gas producer, is equal to item (I, 1), equation (32), of the heat balance. H_2 is the same as item (II, 1), equation (36), of the heat balance, while H_3 is the sum of items (II, 2), equations (37) or (37a), and (II, 5), equation (39). The computation of the efficiencies from the heat balance sheet is therefore a simple matter.

Since, however, the heat balance is established on the basis of the fuel used in the producer *only*, the above efficiency computations, if based on the heat-balance items referred to, will apply only to a suction-gas plant. In a pressure-gas plant, in which fuel is also used in an auxiliary boiler, the items of the heat balance above enumerated must therefore be modified. In most cases the kind of fuel used for the boiler is the same as that used for the producer and this will be assumed here. Then H_1 is the same as before, but for the computation of H_2 and H_3, items (II, 1), (II, 2), and (II, 5) must be modified because, on account of the greater amount of fuel now entering the computation, the yield of gas per unit weight of fuel is less. The new values of H_2 and H_3 can be found by multiplying the items mentioned by the ratio

$$\frac{\text{coal used in producer}}{\text{coal used in producer} + \text{coal used in boiler}}.$$

The "*figure of merit*" is a term invented to compare the performances of various producers or of the same producer under different conditions. It is defined as the heat value of the gas produced per unit weight of carbon it contains, and is computed by dividing the heat value per standard cubic foot of the gas by the weight of carbon the standard cubic foot contains. Wyer gives the following computation as an example:

	Component of Gas, Volume Per Cent.	Heat Value Due to the Various Combustible Components, B.t.u.	Weight of Carbon Represented by the Various Component Gases, Lbs.
CO_2	4.0		.00134
CO	25.4	86.86	.00848
CH_4	1.5	16.05	.00050
H_2	11.1	38.40	
N_2	58.0		
	100.0	141.31	.01032

Hence figure of merit $= \dfrac{141.31}{.01032} = 13{,}690$ B.t.u. per pound of carbon.

394. The Testing of Gas Engines. — Gas engines, like steam engines, may be tested for a variety of purposes. The ordinary test consists of the determination of I.H.P. and B.H.P., and of fuel consumption. If this is carried out for a series of loads, the test is practically one for efficiency and capacity. These are the usual items covered in contract guarantees. There are, however, a number of other things which may serve as the objects of a test on a gas engine, as for instance, the interrelation between economy and speed, economy and proportion of fuel mixture, capacity and proportion of fuel mixture, economy and capacity with variation in jacket temperatures, speed regulation tests,* and a number of others. The important thing, of course, is to keep the ultimate object clearly in mind and to make the arrangements of the test to meet this object.

In the following directions it is assumed that it is desired to make a complete capacity and economy test, from the computed results of which it will be possible to establish a complete heat balance. If in any given practical case the aim of the test is different, changes from the scheme can be easily made and will readily present themselves.

1. *Examine the engine* externally and internally, especially with reference to the condition of valve and piston rings, and take note of all the conditions that may bear on the object of the test. Furnish drawings or sketches of any essential parts that cannot be clearly described in words alone.

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* For speed regulation tests, see Chap. XVIII.

If the test is to be for maximum efficiency and capacity, all defects should be remedied before the tests. In case of a service test, the work is, of course, done with the engine as found, unless the conditions should be clearly abnormal.

2. *Determine the principal dimensions* of the engine — cylinder diameter or diameters, length of stroke, etc. This should be done in all cases, whether dimensions are already known or not, and the cylinder should be hot when the measurement is taken. Find the clearance volume. For the method of doing this, see the Steam Engine Test Code.

3. *Calibrate all the instruments used* — indicators, gauges, water-meters, etc., — if possible, both before and after the test. For methods of doing this, see previous chapters in this book.

4. *Starting and Stopping a Test.* — In tests to determine the maximum efficiency or economy of an engine always operate the machine for a sufficient length of time, as nearly as possible under test conditions, to make sure that all conditions have become fairly constant. Then start the observations and continue for the time decided upon under (5) below.

If the test is to determine the performance under working conditions, the test should start when the engine starts, and should continue to the close of the period covering the day's work.

5. *Duration of Tests.*—The length of the test depends somewhat upon the object of the test. In the case of economy or efficiency tests on an engine using liquid fuel or a " ready-made " gas, like natural gas or illuminating gas, the time of test, after constant conditions have been reached, can be made comparatively short, and may in fact be stopped after several successive readings, say one-half hour apart, have shown that power developed and fuel consumed are fairly constant. Tests on small and very high-speed liquid fuel engines, like automobile engines, sometimes last less than one hour for each load, the reading being taken every 5 or 10 minutes.

Where engines are tested in connection with their gas producers, and the guarantee is based upon the coal consumption, as it usually is, the test should last not less than 12 hours and should preferably continue for more than 24, on account of the difficulty of determining the total fuel consumption.

In case of service tests (plant tests) the test, of course, continues for the usual period of time covering the hours of work in the particular plant.

6. *Apparatus Required, Observations, and Calculations.* — Means must be furnished to determine the following items:

(*a*) Indicated horse power. The factors determining this quantity are the mean-effective pressure in the cylinder, and the number of revolutions or of explosions.

The pressures and rotative speeds encountered in gas-engine work are generally much higher than those found in steam-engine practice. This calls for compact indicators with moving parts as light as possible consistent with stiffness, the use of strong springs, and the substitution of the half-size piston for the normal one. On account of the high temperature of the working medium, external spring indicators are of special service, although many of them labor under the disadvantage of having rather heavy moving parts.

Engines operating at speeds exceeding 500 R.P.M. (auto engines will run up to 1500 or 2000 R.P.M.) either are not indicated at all, or some form of optical indicator (see Chap. XV) must be employed. The indicator pipe must be as short as possible and without sharp bends. This is for the purpose of preventing the accumulation of an explosive mixture in the piping and to reduce pressure loss as far as possible.

The reducing motion may be of any type best adapted to the speed. See Chap. XV.

The number of diagrams to be taken varies with the kind of engine and largely also with the kind of service under which the engine is operating. For an engine which governs by regulating the size of the charge, i.e., in which there are no misstrokes, and where the load is fairly constant, a single diagram every 10 or 15 minutes is quite sufficient. If the load varies, the diagrams may either be taken oftener, or a bundle of diagrams may be taken on the same card. Where the load is very fluctuating, a good method is to operate the indicator for 2 minutes and to apply the pencil to the paper at successive intervals of 10 seconds during that period. The average of all the diagrams so obtained is, of course, used for the determination of I.H.P.

In an engine governing on the hit-and-miss principle, the successive cycles occurring between any two miss periods are usually not at all of the same size. In such a case, therefore, the pencil should be allowed to trace all of the " hit " diagrams between two successive miss periods, the average being used for the computation.

In a multicylinder or double-acting engine, one indicator must be furnished for each cylinder end. The use of a three-way cock, even where this is possible, is not permissible.

The general horse power formula is

$$\text{I.H.P.} = \frac{plax}{33{,}000}, \tag{44}$$

where, as before, p = mean effective pressure per sq. in. of piston
l = length of stroke in feet, and
a = area of piston in sq. in.

x is the number of impulses (explosions) per minute for each operating-cylinder end, and it should be pointed out that this is not necessarily the number of revolutions per minute in a single-acting 2-cycle machine, or one-half this number in a single-acting 4-cycle engine. It would be equal to these numbers if the engine governs by any of the so-called " precision " methods (throttling or regulating fuel supply), but in an engine governed upon the hit-and-miss principle, x is the number of " hits " per minute and depends upon the load. For the determination of its value in this case, see (c), p. 909.

If the mean effective pressure p is found from the work diagram in a 4-cycle engine (not taking the lower loop area into account), or from the power diagram area in a 2-cycle engine, the I.H.P. computed is the *gross indicated horse power*. In either engine this gross I.H.P. is used in three ways: useful effect (B.H.P.), loss in mechanical friction (*friction horse power*), and loss in the pump work (*fluid friction*), the last being represented either by the area of the lower loop in a 4-cycle diagram, or by the pump card in a 2-cycle engine, assuming, of course, that the engine drives its own pumps. The *net I.H.P.* is equal to the gross I.H.P. less the I.H.P. represented in fluid friction. In a 2-cycle engine, the fluid friction horse power is determined by indicating the pump cylinder or cylinders. The ordinary 4-cycle-engine diagram does not show in the ordinary case a sufficiently clear lower loop to admit of integration. For accurate work, a weak spring card should be taken in this case. In many cases, in tests on 4-cycle engines, the power loss represented by the lower loop is neglected, although it may amount to 3 or 4 per cent of the gross I.H.P.

(*b*) The horse power output. This is determined for a gas engine the same as for a steam engine, and the student is referred to the Steam Engine Code and to Chap. X.

(*c*) Speed and number of explosions. The number of revolutions may be found by means of any of the speed-recording instruments described in Chap. VIII. The number of explosions per minute either bears some definite ratio to the R.P.M., in engines that do not govern by hit-and-miss, or the number is irregular in engines that are governed that way, as above pointed out. The number of explosions in such a case usually can be found by attaching a stroke counter to any reciprocating part of the valve gear which fails to move every time the engine makes a misstroke. Another method, and one which also gives a history of the force of the explosions, is to use an *explosion recorder* of the type designed by Mathot. This instrument is shown in Fig. 550. The following is the description of its method of operation, given by the inventor in his book on "Modern Gas Engines and Gas Producer Plants."

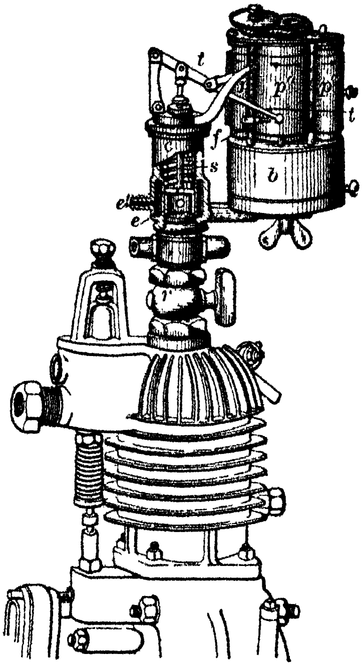

FIG. 550. — MATHOT EXPLOSION RECORDER.

The instrument is somewhat similar in form to the ordinary indicator. Its record, however, is made on a paper tape which is continuously unwound. The cylinder *c* is provided with a piston, about the stem of which a spring *s* is coiled. A clock train contained in the chamber *b* unwinds the strip of paper from the roll *p* and draws it over the drum p'', where the pencil *t* leaves the mark. The tape is then rewound on the spindle p'. A small stylus or pencil *f* traces the atmospheric line on the paper as it passes over the drum p''. In order to obviate the binding of the piston when subjected to the high temperature of the explosions, the cylinder *c* is provided with a casing *e* in which water is circulated by means of a small rubber tube which fits over the nipple e'. This recorder analyzes with absolute precision the work of all engines, whatever may be their speed. It gives a continuous, graphic record from which the number of explosions, together with the initial pressure of each, can be determined, and the order of their succession. Consequently the regularity or irregularity of the variations can be observed and traced to the secondary influences producing them, such as the action of the inlet and outlet valves and the sensitiveness of the governor. It renders it possible to estimate the resistance to suction and the back pressure due to expelling the burnt gases, the chief causes of loss in efficiency in high-speed engines. Furthermore, the influence of compression is markedly shown from the diagram obtained.

The recorder is mounted on the engine; its piston is driven back by each of the explosions to a height corresponding with their force; and the stylus or pencil controlled by the lever *t* records them side by side on the moving strip of paper. The speed with which this strip is unwound conforms with the number of revolutions of the engine to be tested, so that the records of the explosions are placed side by side clearly and legibly.

Their succession indicates not only the number of explosions and of revolutions which occur in a given time, but also their regularity, the number of misfires. The pressure of the explosions is measured by a scale connected with the recorder spring. By employing a very weak spring which flexes at the bottom simply by the effect of the compression in the engine cylinder, it is possible to ascertain the amount of the resistance to suction and to the exhaust. It is simply sufficient to compare the explosion record with the atmospheric line, traced by the stylus *f*. By means of this apparatus, and of the records which it furnishes, it is possible analytically to regulate the work of an engine, to ascertian the proportion of air, gas, or hydrocarbon which produces the most powerful explosion, to regulate the compression, the speed, the time of ignition, the temperature, and the like.

(*d*) Measurement of the fuel supply. The common means for measuring the fuel supply for gases like illuminating or natural gas is the gas meter. This may also be used for pressure-producer gas. On account of the nature of the demand, which is strongly fluctuating, especially if only one cylinder is in operation, it is always best to install a reservoir of some kind between engine and meter. This may be a gas bag, for a small engine, some form of pressure regulator, or nothing but a tank of sufficient capacity to tone down the pressure fluctuations. Without this precaution the determination of average gas pressure at the meter becomes difficult. Besides the pressure and volume, the temperature of the gas at the meter must also be taken. The volume should be reduced to standard conditions, 14.7 pounds and 32° F., in the report.

The most common method of determining the fuel consumption in a suction-gas plant is to measure the coal used in the producer.

The measurement of liquid fuels is made most simply by supplying the engine from a tank, calibrated and fitted with a gauge glass and scale. As a general thing, however, it is more accurate to weigh the fuel. In that case the level of the oil in the tank is noted at the beginning of a test, and the fuel necessary to maintain the level at the same point, or to bring it back to the same point, is weighed and poured in. Where the fuel consumption is large, and a small error in weighing causes only a very small percentage error, a method that has been used is to put the fuel tank directly upon scales, and to connect the tank by flexible piping to the supply piping of the engine.

Besides the weight of the liquid fuel, temperature and density should also be recorded.

In any engine using a fuel supply for the maintenance of an ignition flame, or for any other purpose, the quantity so used must be taken into account and separately determined if possible.

(*e*) Measurement of air supply. This measurement may be made in two ways: by meter on the air-inlet pipe, or by computation from the exhaust-gas analysis in connection with the fuel analysis. In case a meter is used, the remarks under (*d*) with reference to pressure fluctuations apply. For the method of computing the air supply, see Art. 387.

(*f*) Measurement of jacket water. This may be determined by the use of water meters or of weighing tanks. The latter method is preferred where it can be applied. The arrangement of tanks, and whether the water should be measured before entering or after leaving the jackets, is largely a question of expediency, and must be decided for every particular case.

If an engine uses a separate supply for cylinder jackets, and for piston and rod cooling, each supply must be determined separately.

(*g*) Analysis of fuel and exhaust gases. A complete test should always include the chemical analysis of fuel and exhaust gas. For the methods of analyzing solid fuels and exhaust gases, the student is referred to Chapter XIII. For the analysis of fuel gases, see Art. 386, of this chapter. Methods of sampling, etc., are also discussed in the places referred to.

(*h*) Heating value of the fuel. In the case of solid fuels (gas producer plants), the heating value is determined upon the laboratory sample taken on the test. The same thing may be done in the case of gas fuels, samples being taken as described in Arts. 275 and 386. Whenever possible, however, a gas calorimeter should also form a part of the test apparatus and the determinations of heating value should be made during the test. The computations of heating value based on the laboratory analysis will then serve as a check upon the calorimeter work. In stating gas-heating values in B.t.u. per cubic foot, the volume should be reduced from the conditions of test to *standard conditions*, 14.7 pounds pressure and 32° F.

(*i*) Pressure, temperature, and other observations. The pressure observations required are: pressure of gas at meter and barometer pressure.

The temperature readings are: outside air, air in engine room, inlet and outlet of jacket water, gas at meter, or fuel oil at reservoir, and temperature of exhaust gas. The latter is best taken by means of some reliable pyrometer and as close to the exhaust valve as possible.

Other observations should be: density of fuel oil, if an oil engine is tested, and humidity of atmosphere, by wet and dry bulb thermometer, or by some other form of hygrometer.

7. *Recording the Data, and Preparation of Report.* — The time of taking weights and of recording every observation should be carefully noted in the logs. Note every occurrence on the test. Readings 20 or 30 minutes apart are satisfactory when the conditions are uniform. Where they vary, readings at more frequent intervals should be made. It is good practice to carefully make all readings with special care at the end of every hour, so as to divide the test into hour periods for the purpose of checking uniformity of conditions as the test proceeds. It is also desirable to plot during the test curves of the fuel consumption per hour against brake horse power (similar to Willan's Law, see Chap. XVIII). This will at once reveal any abnormal conditions that may become operative after the test is started.

The following table, published in the Code established by the American Society of Mechanical Engineers, shows the items that should be computed for a complete report.

This is followed by two abridged forms of report, used in Sibley College, one for a gas, the other for an oil engine. These forms give quite sufficient information for the ordinary commercial report and furnish space for the recording of a series of " runs." Where these have been made at a series of loads, curves of efficiency and fuel consumption should be plotted, and should accompany the report, to graphically represent the results. (See type examples on such curves for steam engines in Chap. XVIII).

DATA AND RESULTS OF TEST OF GAS OR OIL ENGINE.

Arranged according to the Complete Form advised by the Engine Test Committee, American Society of Mechanical Engineers. Code of 1902.

1. Made by . . ..of ....
on engine located at..............................
to determine ....
.. . .
2. Date of trial..............................

3. Type of engine, whether oil or gas..................................
....

4. Class of engine (mill, marine, motor for vehicle, pumping, or other).....
....

5. Number of revolutions for one cycle, and class of cycle..................
....

6. Method of ignition..
....

7. Name of builders ....

8. Gas or oil used...

(a) Specific gravity.................................. deg. F.
(b) Burning point..................................... "
(c) Flashing point.................................... "

9. Dimensions of engine:

1st Cyl. 2d Cyl.

(a) Class of cylinder (working or for compressing the charge)..........
(b) Vertical or horizontal..........
(c) Single or double acting..........
(d) Cylinder dimensions..........
Bore.......................... in.
Stroke........................ ft.
Diameter piston rod........... in.
Diameter tail rod............. in.
(e) Compression space or clearance in per cent of volume displaced by piston per stroke...
Head end..........................
Crank end.........................
Average...........................
(f) Surface in square feet (average)..........
Barrel of cylinders ....
Cylinder heads....................
Clearance and ports...............
Ends of piston....................
Piston rod........................
(g) Jacket surfaces or internal surfaces of cylinder heated by jackets, in square feet.........
Barrel of cylinder................
Cylinder heads....................
Clearance and ports...............
(h) Horse-power constant for one pound M.E.P. and one revolution per minute.........

10. Give description of main features of engine and plant, and illustrate with drawings of same given on an appended sheet. Describe method of governing. State whether the conditions were constant throughout the test.

Total Quantities.

11. Duration of test.. hours.
12. Gas or oil consumed.................................... cu. ft. or lbs.
13. Air supplied in cubic feet.............................. cubic feet.
14. Cooling water supplied to jackets...................... cu. ft. or lbs.
15. Calorific value of gas or oil by calorimeter test, determined by ....calorimeter... .... B.t.u.

Hourly Quantities.

16. Gas or oil consumed per hour.......................... cu. ft. or lbs.
17. Cooling water supplied per hour........................ pounds.

Pressures and Temperatures.

18. Pressure at meter (for gas engine) in inches of water.... inches.
19. Barometric pressure, inches of mercury................ “
20. Temperature of cooling water:
    (*a*) Inlet .... deg. F.
    (*b*) Outlet .... “
21. Temperature of gas at meter (for gas engine)........ “
22. Temperature of atmosphere:
    (*a*) Dry-bulb thermometer............................ “
    (*b*) Wet-bulb thermometer............................ “
    (*c*) Degree of humidity.............................. per cent.
23. Temperature of exhaust gases.......................... deg. F.
    How determined..

Data Relating to Heat Measurement.

24. Heat units consumed per hour (pounds of oil or cubic feet of gas per hour multiplied by the total heat of combustion) .... B.t.u.
25. Heat rejected in cooling water:
    (*a*) Total per hour.................................. “
    (*b*) In per cent of heat of combustion of the gas or oil consumed.................................. per cent.
26. Sensible heat rejected in exhaust gases above temperature of inlet air:
    (*a*) Total per hour.................................. B.t.u.
    (*b*) In per cent of heat of combustion of the gas or oil consumed.................................. per cent.

27. Heat lost through incomplete combustion and radiation per hour:
    (*a*) Total per hour.................................... B.t.u.
    (*b*) In per cent of heat of combustion of the gas or oil consumed.. per cent.

Speed, Etc.

28. Revolutions per minute............................... rev.
29. Average number of explosions per minute.....................
    How determined...
30. Variation of speed between no load and full load........ rev.
31. Fluctuation of speed on changing from no load to full load measured by the increase in the revolutions due to the change.

Indicator Diagrams.

	1st Cyl	2d Cyl
32. Pressure in pounds per square inch above atmosphere:		
(*a*) Maximum pressure		
(*b*) Pressure just before ignition....................		
(*c*) Pressure at end of expansion....................		
(*d*) Exhaust pressure		
33. Temperatures in deg. F. computed from diagrams:		
(*a*) Maximum temperature (not necessarily at maximum pressure)............................		
(*b*) Just before ignition..............................		
(*c*) At end of expansion...............................		
(*d*) During exhaust		
34. Mean-effective pressure in pounds per square inch...		

Power.

35. Power as rated by builders:
    (*a*) Indicated horse power............................ H.P
    (*b*) Brake.. "
36. Indicated horse power actually developed:
    First cylinder.. "
    Second cylinder... "
    Total... "
37. Brake H.P., electric H.P., or pump H.P., according to the class of engine... "
38. Friction indicated H.P. from diagrams, with no load on engine and computed for average speed..........
39. Percentage of indicated H.P. lost in friction............ per cent

Standard Efficiency Results.

40. Heat units consumed by the engine per hour:

    (*a*) Per indicated horse power.................... B.t.u.
    (*b*) Per brake horse power.......................... "

41. Heat units consumed by the engine per minute:

    (*a*) Per indicated horse power...................... "
    (*b*) Per brake horse power.......................... "

42. Thermal efficiency ratio:

    (*a*) Per indicated horse power.................... per cent.
    (*b*) Per brake horse power.......................... "

Miscellaneous Efficiency Results.

43. Cubic feet of gas or pounds of oil consumed per H.P. per hour:

    (*a*) Per indicated horse power......................
    (*b*) Per brake horse power..........................

Heat Balance.

44. Quantities given in per cents of the total heat of combustion of the fuel:

    (*a*) Heat equivalent of indicated horse power........ per cent.
    (*b*) Heat rejected in cooling water........... "
    (*c*) Heat rejected in exhaust gases and lost through radiation and incomplete combustion .... "
    Sum = 100 "

    Subdivisions of Item (*c*):

    (*c*1) Heat rejected in exhaust gases.............. "
    (*c*2) Lost through incomplete combustion........... "
    (*c*3) Lost through radiation, and unaccounted for...... "
    Sum = Item (*c*)........ "

Additional Data.

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is to be used. Also give copies of indicator diagrams nearest the mean and the corresponding scales. Where analyses are made of the gas or oil used as fuel, or of the exhaust gases, the results may be given in a separate table.

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Data and Results of Test of . . . . *Gas Engine*

By *190*

Object of Test . . . .

Dimensions of Engine.

Rated H.P at . . . . R.P.M. =
Diameter of piston . . . . in.
Area of piston . . . . . . sq. in
Length of stroke. . . . . ft
Piston displacement. . . . . cu. ft
Clearance . . . . cu ft
" . . . . per cent
Diameter piston rod. . . . . in
" crank pin. . . . . . in
Scale of indicator spring . lbs. per in

Data

Run No	I	II	III	IV.
Duration trial, hrs				
Brake load, net lbs				
Gas, total cu ft.				
* Gas per hour, cu. ft.				
Air, total cu ft				
* Air per hour, cu ft.				
Ratio air to gas by weight.				
Jacket water, total lbs				
" per hour, lbs				
" tempt. entering, F°.				
" " leaving, F°				
" range, F°.				
Revolutions, total				
" per hour				
" per min.				
Cycles, per min				
Explosions, total				
" per hour				
" per min.				
Ratio of explosions to cycles				
Temperature, exhaust, F°				
" room F°				
" range				
* Gas, wt. of a cu. ft. lbs				
* Air, wt of a cu ft. lbs				
* Mixture, wt of a cu ft., lbs				
Specific heat, gas				
" air				
" exhaust gases				
* Thermal equiv., cu. ft. gas, B.t u.				

Run No.	I.	II	III.	IV.
Results.				
INDICATOR.				
Maximum press. lbs. sq. in.				
Compression press. lbs sq. in.				
M E P power stroke.				
" comp. "				
I.H.P. net				
D.H.P				
Friction horse power				
Mechanical efficiency, per cent				
Weight of gas per hr., lbs.				
Weight of air per hr., lbs.				
* Gas per I H.P, per hr, cu ft				
" " " lbs.				
" D.H.P., " cu ft.				
" " " lbs.				
Heat per hour.				
Supplied . . B t u.				
" per cent				
Absorbed by jacket water . B t u				
" " " " per cent				
Exhausted . B.t.u				
" per cent				
Thermal equiv ind work . . B.t u				
" " " " . . per cent				
Radiation and loss . . . B.t u				
" " . . per cent				
Thermal units per I H P., per hr.				
" " " D.H P., " "				
EFFICIENCIES.				
Thermal from I H P . . per cent				
" " D H.P. . . . per cent				
Volumetric Efficiency. . . . per cent				

---

* At 32° F. and 14.7 lbs. absolute pressure per sq. in.

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Data and Results of Test of .... *Oil Engine*

Observers { ....
.... *Date*

Dimensions of Engine.		
Rated H.P. at	R.P.M. =	
Diameter of piston	in.	
Area of piston	sq. in	
Length of stroke	ft.	
Piston displacement	cu. ft.	
Clearance	cu. ft	
"	per cent	
Length of brake arm		
Scale of indicator spring	lbs. per in.	

Data.

Run No	I.	II	III	IV
Duration trial, hrs.				
Brake load, net lbs.				
Oil, total cu. in.				
Oil per hour, cu in				
Air, total cu. ft, meter				
* Air per hour, cu ft., standard				
Weight of air per hr., lbs.				
Ratio air to oil by weight				
Weight of ex gases per hr, lbs				
Jacket water, total lbs				
" per hour, lbs				
" tempt entering, F°				
" " leaving, F°				
" range, F°				
Revolutions, per hour				
" per min				
Cycles, per min				
Explosions, per hour				
" per min				
Ratio of explosions to cycles				
Temperature, exhaust, F°				
" air, F°				
" range, F°				
Specific gravity of oil				
* Air, wt. of a cu ft, lbs				
Specific heat, exhaust gases				
Heat in one lb oil, B.t.u				

Run No.		I.	II.	III	IV
Results.					
INDICATOR AND POWER.					
Maximum press, lbs, sq in					
Compression press, lbs sq. in					
M.E.P., net					
I.H.P., net					
D.H.P.					
Friction horse power					
FUEL CONSUMPTION.					
Oil, per hour, pints					
" " " lbs					
Oil per I.H.P per hr, pints					
" " " lbs.					
" D.H.P. " pints					
" " " lbs					
HEAT PER HOUR.					
Supplied	B t u				
"	per cent				
Thermal equiv. ind work	B t u				
" " " "	per cent				
Absorbed by jacket water	B t u				
" " " "	per cent				
Exhausted	B t u				
"	per cent				
Radiation and loss	B t u.				
" "	per cent				
Thermal units per I H P per hr.					
" " " D H P "					
EFFICIENCIES					
Mechanical efficiency	per cent				
Thermal from I H P	per cent				
" " D H P.	per cent				
Volumetric efficiency	per cent				

* At standard temperature and pressure (32° F. and 14 7 lbs. abs.)

395. The Heat Balance for an Internal Combustion Engine. — The items of a complete heat balance for an internal combustion engine may be tabulated as follows:

	B.t.u.	Per cent
(*a*) Heat supplied per hour	.....	100.00
(*b*) Heat equivalent of I.H.P.-hour	.....	..... .
(*c*) Heat rejected in cooling water per hour..	......	......
(*d*) Sensible heat lost in exhaust per hour ...	......	......
(*e*) Heat lost in exhaust through incomplete combustion per hour	......	......
(*f*) Radiation, conduction, etc., per hour....	......	......

(*a*) The heat supplied per hour is the product of the quantity of fuel used in that time multiplied by the heating value of unit quantity of that fuel. There is some disagreement concerning which of the heating values, the higher or the lower, should be used. American practice seems to sanction the use of the higher value, on the ground that the lower heating value is an indefinite quantity, varying with conditions.

In a gas-engine test care should be taken to see that the heating value of the gas per cubic foot is stated for the same pressure and temperature conditions as the hourly consumption.

It is assumed that the heat quantities in the balance are computed above some reference temperature, ordinarily taken as that of the room. That being the case, the heat energy in the fuel is the only heat supplied to the engine, provided, of course, that the fuel is initially at room temperature.

(*b*) The heat equivalent of the indicated horse-power per hour

$$= (\text{I.H.P.} \times 2545)\ \text{B.t.u.} \qquad (45)$$

(*c*) The heat rejected in cooling water per hour

$$= W\,(t_2 - t_1)\ \text{B.t.u.} \qquad (46)$$

in which W = weight of cooling water per hour,

t_2 = outlet temperature of water,

and t_1 = inlet temperature.

In case cooling by vaporization is used, the heat rejected will be approximately

$$= W(970 + 212 - t_1),$$
$$= W(1182 - t_1) \text{ B.t.u.}, \tag{46a}$$

in which W = weight evaporated per hour.
970 = latent heat of evaporation under atmospheric pressure.
212 = approximate atmospheric boiling temperature,
and t_1 = temperature of cooling water entering the jacket.

The reason that this expression is not quite correct lies in the fact that some vaporization takes place at a temperature less than 212°, depending upon barometer pressure and humidity conditions in the surrounding air.

(*d*) and (*e*) For the computation of the heat losses in exhaust, see Art. 387, in this chapter.

(*f*) The radiation loss is determined from the equation

$$f = a - (b + c + d + e). \tag{47}$$

396. The Computation of Gas-Engine Efficiencies. — (*a*) The cylinder efficiency, E_c. This efficiency is the ratio of the area of the real average indicator card obtained on a test divided by the area of the theoretical diagram that would have been obtained with the same charge, but provided that there had been no losses of any kind.

The efficiency of this theoretical cycle, which for an Otto engine consists of an adiabatic-compression and an adiabatic-expansion line combined with a constant-volume combustion and a constant-volume discharge line, has already been computed on the basis of temperatures on p. 349. Now with the information at hand concerning the composition and weight of charge, it becomes possible to compute the data for the construction of such a theoretical cycle by means of the laws laid down in Chapter XI, *provided that the temperature at one point is known.* Assuming that this is the case (see the assumption usually made, as explained in connection with the construction of the entropy diagram, Chapter XI), the theoretical diagram found will bear a relation to the real indicator card (shaded area) approximately as shown in Fig. 551. It should be pointed out that to make this work as accurate as possible, variation of specific heat with temperature should be taken into account, which makes the computation far from simple. The cylinder efficiency is then equal to

$$E_c = \frac{\text{area } aebcda}{\text{area } a'b'c'd'a'} . \qquad (48)$$

Another way of computing E_c is to divide the thermal efficiency based on I.H P. (see item (*b*) below) by the efficiency of the theoretical cycle, which, from equation 73, p. 349, for this cycle is $1 - \frac{T_2}{T_1}$, where T_2 = absolute temperature at a', and T_1 absolute temperature at b'.

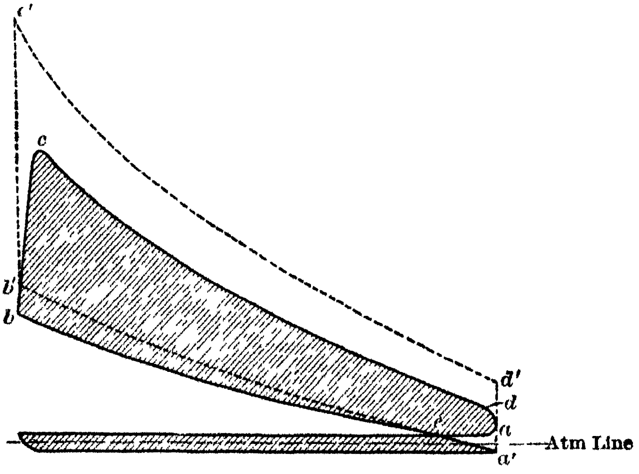

Fig 551

The cylinder efficiency is an expression of the degree to which the real engine approaches the performance of the ideal, and gives a good idea of the magnitude of the sum of the losses in the real engine.

(*b*) The thermal efficiency, E_t. This may be computed on the basis of either I.H.P. or B.H.P. If H represents the heat supplied per hour (item (*a*) of the heat balance in Art. 395), then

$$\text{Thermal efficiency on I.H.P.} = \frac{\text{I.H.P.} \times 2545}{H}, \qquad (49)$$

and $$\text{Thermal efficiency on B.H.P.} = \frac{\text{B.H.P.} \times 2545}{H}. \qquad (50)$$

It is becoming the general practice of engineers to express engine economy in terms of B.t.u. per horse power per hour instead of cubic feet of gas or pounds of oil. The thermal efficiencies then may be computed simply by dividing 2545 by the heat consumption per I.H.P. or per B.H.P., as the case may be.

(*c*) The mechanical efficiency, E_m. This is the ratio of brake horse power to indicated horse power $\left(\frac{\text{B.H.P.}}{\text{I.H.P.}}\right)$. Some difference of opinion exists among engineers as to what constitutes the value of the I.H.P. factor in this ratio. As was pointed out in Art. 394, item 6 (*a*), we distinguish a gross and a net I.H.P. in gas engines, the difference between them being the fluid friction, measured in a 4-cycle engine by the area of the lower loop, in a 2-cycle engine by the pump work. There is no doubt that, if the mechanical efficiency is to express the real mechanical loss in friction, the formula should read

$$E_m = \frac{\text{B.H.P.}}{\text{I.H.P.}_{\text{net}}}. \qquad (51)$$

If, on the other hand, it is desired to get an idea of the commercial efficiency of the machine for turning I.H.P. into B.H.P., the formula should read

$$E_m = \frac{\text{B.H.P.}}{\text{I.H.P.}_{\text{gross}}}. \qquad (52)$$

In any given test of importance, the mechanical efficiency should be stated both ways in the report.

(*d*) The volumetric efficiency, E_v. This efficiency is the ratio of the total volume of the fuel mixture taken into the cylinder under standard pressure and temperature conditions, per suction stroke, divided by the piston displacement per suction stroke. The charge at the end of the suction stroke is under a comparatively high temperature, having been heated by the cylinder walls, and is further at a pressure slightly below atmosphere. Both of these factors combine to decrease the *charge weight* per cycle, upon which engine capacity directly depends, and the volumetric efficiency is therefore in a sense a measure of how nearly theoretical engine capacity is realized.

This efficiency can be found accurately only if both fuel and air supply have been determined on a test. Then if

V = number of cubic feet of gas used per hour, under standard conditions,
r = volume ratio of air to gas,
D = diameter of cylinder in feet,
L = length of stroke in feet, and
x = number of suction strokes per minute,

we will have, for a 4-cycle engine,

$$E_v = \frac{\dfrac{V + Vr}{60\,x}}{\dfrac{\pi D^2}{4}L} = \frac{V + Vr}{47.1\,D^2Lx}. \tag{53}$$

In case of an engine using a liquid fuel, the computation becomes uncertain because of the fact that the volume of the liquid fuel vapor is a function of cylinder temperature.

The volumetric efficiency may be approximately computed directly from a lower loop card. Let Fig. 552 represent the lower loop on a 4-cycle card. The piston displacement is measured by the distance a. The volume of the mixture in cylinder under atmospheric pressure is measured by the distance b. Hence volumetric efficiency is approximately equal to $\frac{b}{a}\cdot$ The computation is not exact because the temperature factor is left out of account.

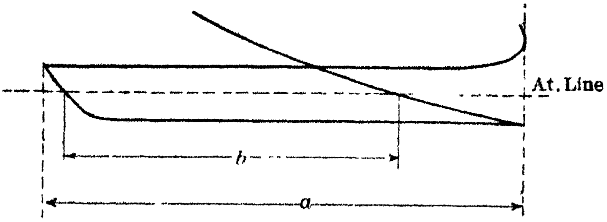

FIG. 552.

CHAPTER XXII.

HOT-AIR ENGINES

397. Types of Engines and Methods of Operation. — Hot-air engines belong to the class of external combustion engines, as distinguished from ordinary gas engines, which are true internal combustion engines. The difference lies in the fact that in external combustion engines the working medium, which is usually air, receives its stock of heat for conversion into work from an external source. As stated, a part of this stock of heat is converted into work, the rest is discharged by cooling the working fluid by an external source. No actual discharge of working fluid, except for leakage, takes place, the same body of fluid serving for every cycle. In gas engines, on the other hand, a new charge of working fluid is taken in every cycle, and the heat supplied for conversion into work is chemically generated by the process of combustion in the cylinder itself during a certain part of the cycle. The heat remaining at the end of each cycle is discharged with the working fluid.

There are at present only two types of hot-air engines in the market, the Ericsson and the Rider. These are commonly used as pumping engines. Their capacity is generally small, owing mainly to two causes. In the first place, air used as a working medium requires large cylinder capacities per unit of power as compared with those required by other media, like steam or fuel gas. This is especially true if it should be attempted to use the Carnot cycle. Neither the Ericsson nor the Rider engine operates on this cycle, even in theory, but the disadvantage stated still exists. In the second place, it is not easy actually to construct large engines of this type with a satisfactory furnace in which the heat can be generated and transferred. For the same reason the thermal efficiency of even the small machines is in practice generally low. Nevertheless, they have certain advantages where small power is all that is required, and there is no skilled attendance available, since, in order to operate them, it is merely necessary to light a coal or gas fire in the furnace.

398. The Ericsson Engine. — Fig. 553 gives a clear idea of the construction of this engine. The cylinder 1 contains two pistons, —

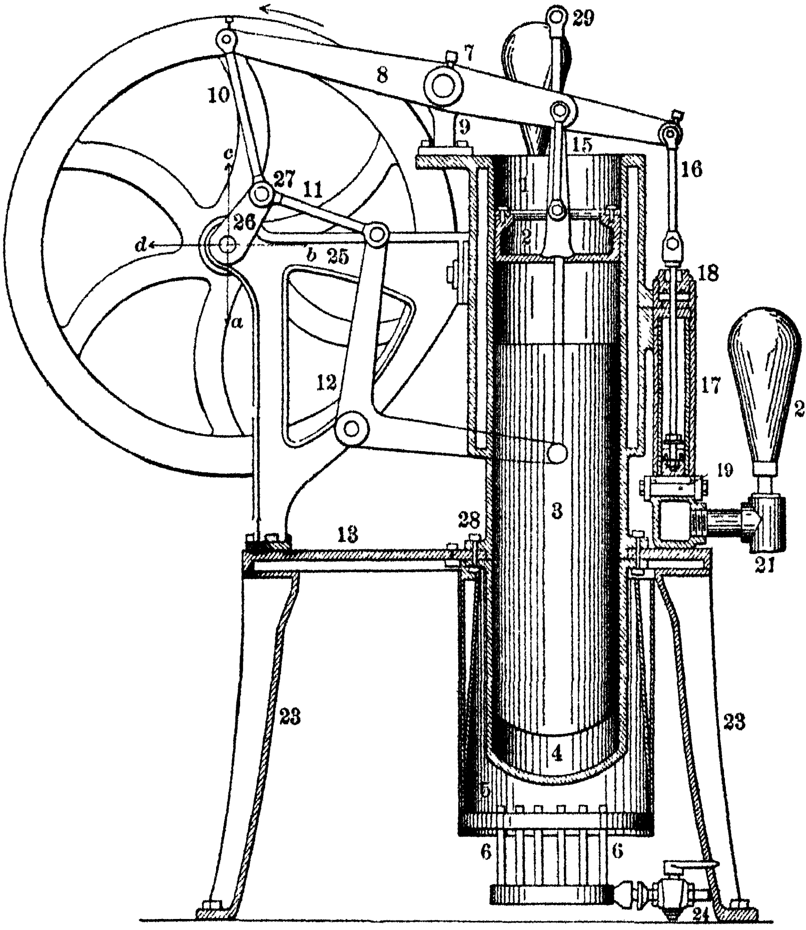

Fig. 553. — Ericsson Hot-air Engine.

the disk piston 2, called the air or power piston, and the transfer piston 3. The power piston 2 is connected through rod 15 to the main beam 8. At one end the main beam operates through rod 16 the single-acting plunger pump 17. This pump lifts the water on the up stroke and forces it through the jacket surrounding the upper part of cylinder 1 and into the discharge main. This jacket constitutes the *cooler* for the working medium. On the other end the main beam through rod 10 operates the crank 26, to which in turn are connected the rod 11 and the bell crank 12. How this motion is transmitted from the bell crank to the transfer piston 3, by means of side rods, is best shown in the small elevation, Fig. 554.

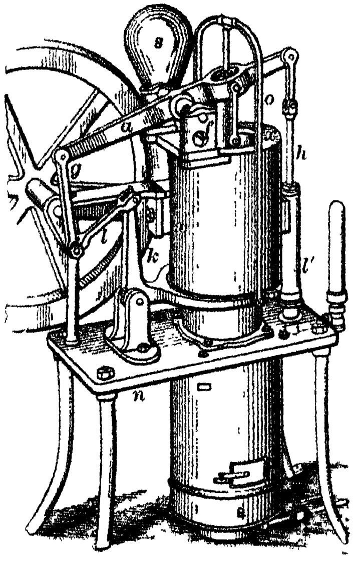

FIG. 554. — ERICSSON ENGINE.

The transfer piston 3, Fig. 553, is made of light metal and is hollow, acting as a non-conductor. The space 4 is the *heater*, heat being in this case furnished by a set, 6, of Bunsen gas burners. The space 5 is called the furnace.

The method of operation of this engine is best explained by reference to Fig. 555, which represents curves of the movements of the two pistons for one complete turn of the main crank. The link motion is drawn for position 1, with the main crank vertical. Motion is counter-clockwise. The other crank positions are taken at 2, 3, and 4, as shown. The corresponding piston positions are located on vertical lines marked with the same numbers. Connecting the 5 points (since position 1 is repeated at the right) so located gives the curve AB for the motion of the power piston and the curve CD for that of the transfer or displacement piston. Curve $C'D'$, parallel to CD at all points, follows the motion of the lower end of the transfer piston, showing the volumes contained between it and the bottom of the heater, which is indicated by line EF.

In following out the pressure and volume changes in the body of working fluid in this engine, it is necessary to remember (*a*) that only the working piston can change the volume, and (*b*) that the temperature is controlled by the action of the transfer piston.

The changes can now be outlined as follows:

1st *Quadrant* (1 to 2). Power piston rises, causing expansion. Transfer piston rises, forcing air into heater. Heating effect apparently strong enough to overcome pressure drop due to expansion, and net result is fairly constant pressure. Line 1–2 in Fig. 556, which represents an actual indicator diagram from an Ericsson engine. (Full size, scale of spring 10 pounds.)

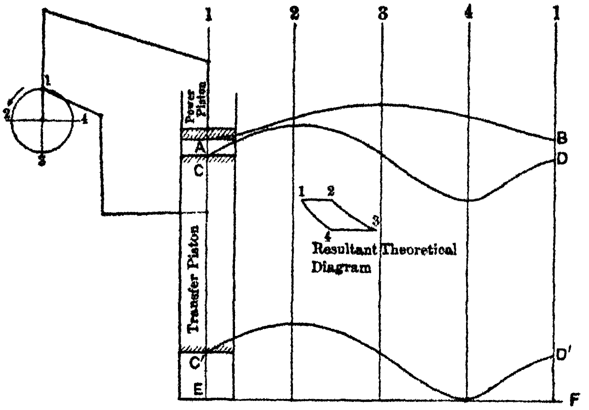

FIG. 555.—DIAGRAM OF OPERATION OF ERICSSON ENGINE.

2d *Quadrant* (2 to 3). Power piston completes rise, causing expansion. Descent of transfer piston forces air into cooler, lowering temperature. The net result is a pressure drop, forming line 2–3 on the card, Fig. 556.

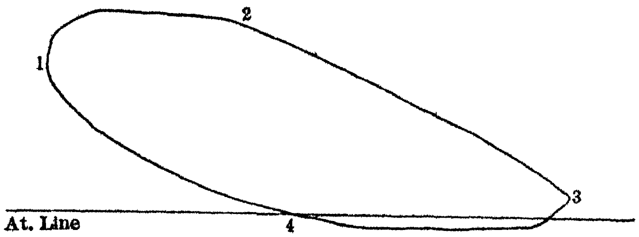

FIG. 556.—INDICATOR DIAGRAM FROM ERICSSON ENGINE.

3d *Quadrant* (3 to 4). Power piston starts to descend, causing compression. But at the same time the further descent of the transfer piston, forcing more air into the cooler, intensifies the cooling action to such an extent as to practically maintain constant pressure. Line 3-4 on the card.

4th Quadrant (4 to 1). Power piston completes descent, causing compression. Transfer piston rises, forcing air into the heater, raising the temperature. The net result is a rise in pressure, forming line 4-1 on the indicator card.

The theoretical cycle upon which the engine operates (two constant pressure lines crossed by two isothermals) and to which Fig. 556 approximates is discussed in Art. 185, p. 348.

399. The Rider Engine.—Fig. 557 shows the construction of this machine. The essentials are: a power piston *D* and a transfer piston *C*, rigidly connected by cranks to the shaft *II*. The two cylinders in which these pistons are working are connected across by the passage *HH* which forms the ***regenerator.*** The regenerator chamber is partly filled with some loosely packed material (sheet steel) which serves to abstract a part of the heat from the working fluid as it flows from the heater to the cooler, and restores this heat when the flow reverses. Below the power cylinder *B* is located a furnace *F*. In this case the fuel is coal, although of course any source of heat may be employed. The work done in the power cylinder is expended in the transfer cylinder and in the pump which is operated from the transfer piston, as shown. The water pumped is first forced through a jacket *E*, serving as a cooler for the transfer cylinder.

To follow the pressure and volume changes which occur in the working fluid during one complete turn, see Fig. 558. Crank positions 1, 2, 3, and 4 represent those of the power piston. The transfer crank has somewhat smaller throw than the power crank and lags behind the latter a few degrees over 90. This gives 1′, 2′, 3′, and 4′ as the corresponding positions of the transfer crank. Although in the drawing the transfer piston and cylinder should be located directly behind the power cylinder, it has been drawn to one side for the sake of clearness. Curve *AB* represents the motion of the lower end of the power piston and line *CD* the bottom of the power cylinder (heater). Similarly, *EF* shows the motion of the lower end of the transfer piston and line *GH* the bottom of the transfer cylinder (cooler). Vertical distances between *AB* and *CD*,

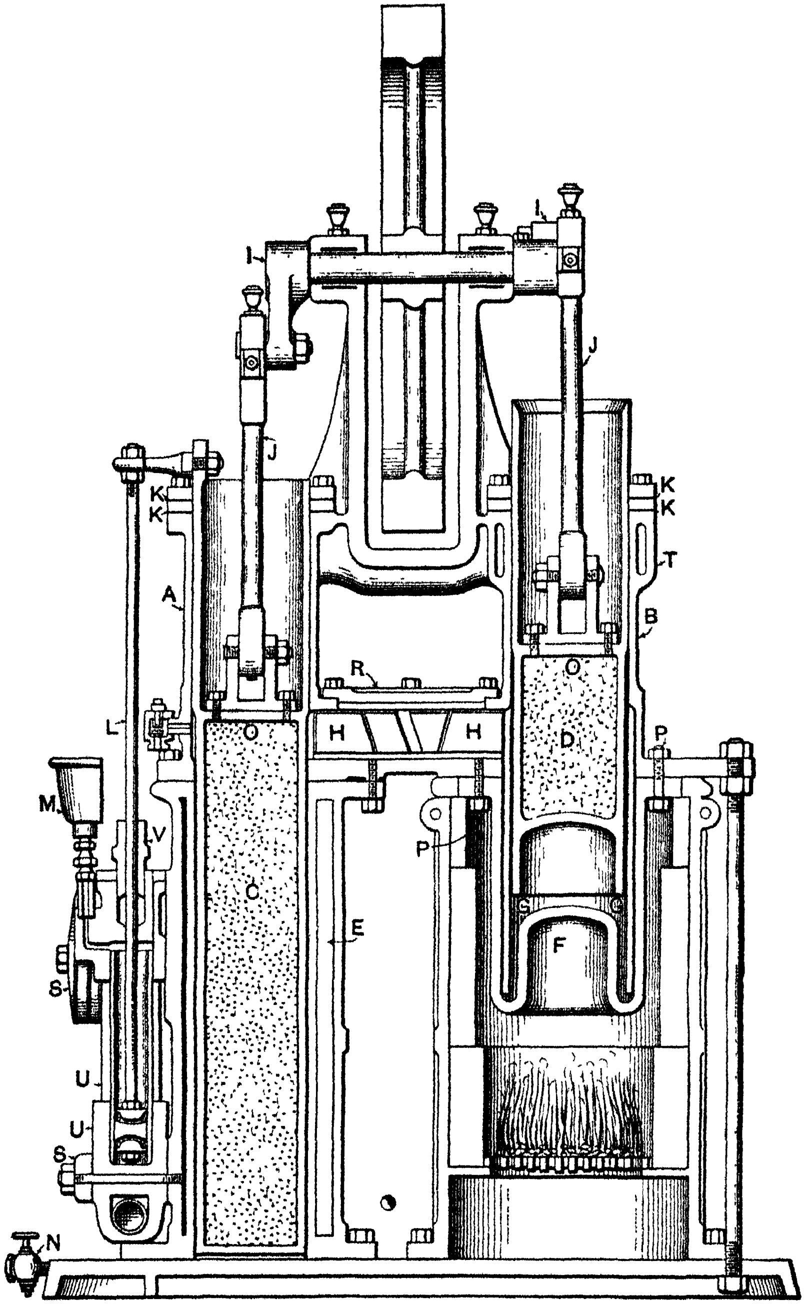

Fig. 557.—Rider Hot-air Engine.

and between *EF* and *GH*, represent at any instant the volumes in the two cylinders. Curve *IJ* shows the sum of the volumes in the two cylinders. This is not the total volume of the working fluid as the regenerator volume *HH*, Fig. 557, is not accounted for; but curve *IJ* will serve to show volume changes, as the regenerator volume is constant. (The regenerator passage is at no time closed by either piston.)

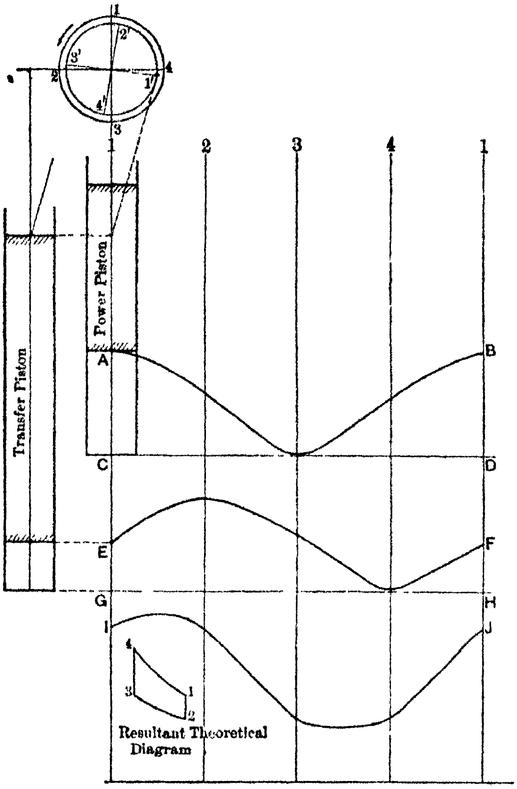

FIG. 558.—DIAGRAM OF OPERATION OF RIDER ENGINE.

1st Quadrant (1 to 2). Practically constant volume change. Descent of power piston and rise of transfer piston indicates abstraction of heat by regenerator and cooling in the transfer cylinder. The net result is a constant volume pressure drop. (See line 1–2 in the theoretical line drawn in Fig. 558.)

2d Quadrant (2 to 3). The volume curve *IJ* shows decided compression. The power piston completes its descent, but the transfer piston also starts to fall, so that the temperature changes are less pronounced than in the first quadrant. The net result is a rise in pressure with volume decrease. (Line 2–3 in diagram in Fig. 558.)

3d Quadrant (3 to 4). Practically constant volume change, accompanied by a rise in pressure as the working fluid is transferred across the regenerator into the heater by the further descent of the transfer piston and the ensuing rise of the power piston. (Line 3–4 in the diagram in Fig. 558.)

4th Quadrant (4 to 1). Marked expansion according to curve *IJ*. This expansion is accompanied by smaller movement of working fluid, on account of simultaneous rise of both pistons. Hence, the heating effect is less strong than in the third quadrant and the net result is expansion with pressure drop. (Line 4–1 in the diagram in Fig 558.)

It will appear from this analysis that the theoretical cycle to which the series of changes approximates is one composed of two constant-volume lines crossed by two isothermals. This is the Stirling (regenerator cycle) which is discussed in Art. 185, p. 348.

In actual testing practice, two indicator diagrams are taken from this engine. The single indicator used is connected to the regenerater chamber and is connected in turn with two reducing motions, the first of which gives the proper reduction for the power piston, the second for the transfer piston. Fig. 559 shows actual diagrams taken from a Rider engine (full size, scale of spring 10 pounds). The methods by which they are traced can be easily followed by noting the periods of piston movements, that is, outstroke and instroke, from curves *AB* and *EF*, Fig 558, and combining with this the knowledge concerning pressure changes gained from the analysis above. Thus for the curve *AB* of the power piston, the first and second quadrants represent the "down" or "in" stroke. From the analysis above, the first quadrant is accompanied by a pressure drop, the second quadrant by a pressure rise. Hence line 1–2–3, Fig. 559, is evidently the one traced during these two periods. The other lines may be identified in a similar manner.

The larger area is that for the power piston, the smaller one that for the transfer piston. The lines are traced in opposite directions (see arrows), as can be discovered if the method of tracing the lines above indicated is fully carried out. The larger area shows work developed in the power cylinder; the smaller, work done upon the transfer piston. Hence the *difference* between the power represented by these two cards is the net power developed by the engine and is that available for pumping and for overcoming mechanical friction.

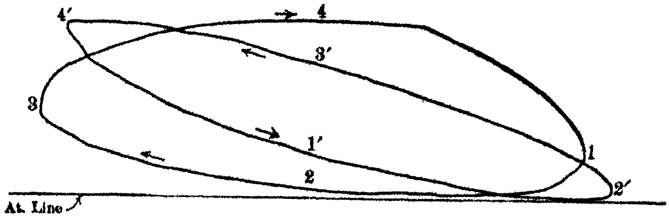

FIG. 559.—INDICATOR DIAGRAMS FROM RIDER ENGINE.

400. **The Testing of Hot-air Engines.**—A complete test of a hot-air engine should include the following items:

(*a*) *The indicated horse power.* The indicator can be operated directly from the pump rod in the Ericsson engine without the interposition of any reducing motion. In the Rider engine, motion is taken from the pistons, but on account of the length of stroke some type of reducing motion must be used. The maximum pressures are in either case low, a 10-pound spring being satisfactory in most cases.

(*b*) *The pump horse power.* On account of the small size of the pump cylinders no pump diagrams need be taken, the useful output being computed directly from the water pumped and the head pumped against. The quantity of water may be determined by weighing, although a nozzle may be used to advantage. The head pumped against is of course the sum of suction and discharge heads with proper allowance for velocity heads (see Chapter XVIII, p. 774).

(*c*) *Measurement of the fuel used.* Where gas is used, the best means is of course the gas meter. In case coal is fired, the test should be extended over a considerable period of time. In the case of gas, a trial lasting one hour for each set of conditions, after the latter have become constant, is satisfactory; but in the case of solid fuel a duration of test of 6 or 8 hours is none too long, on account of the errors involved in judging the beginning and end conditions of the fuel bed on the grate.

(*d*) *Analysis of the fuel with determination of heating value.* This is necessary for the computation of efficiency and the establishment of heat balance. (See Chapters XIII and XXI.)

(*e*) *Analysis of the waste gases from the furnace, together with a determination of their quantity.* Required where it is desired to establish a fully itemized heat balance. (See Chapters XIII and XXI.)

(*f*) *Other observations are:* Suction head; discharge head; revolutions of fly-wheel shaft; range of temperature of jacket (cooler water); pressure in gas mains, if gas is used; temperature of room; barometer reading; dimensions of engine.

The usual method of testing is to divide the total head against which the engine can pump into a number of equal parts, 5 or 6, and at each one of the heads to make a trial run, maintaining the head constant and adjusting the fuel supply to maintain the speed. The following blanks show two convenient forms, the first of which is for recording data, the second for recording results.

401. Computations and Report. — These are comparatively simple and need little comment.

The mechanical efficiency in the case of the Rider engine is the ratio between the pump horse power and the net indicated horse power.

Thermal efficiency may be computed on the basis of indicated work or of pump work. In the latter case it is the ratio of the heat equivalent of the pump work done in a given time, divided by the total heat above room temperature in the fuel supplied in the same time. On the other basis, the numerator of the ratio is the heat equivalent of the indicated work. The former efficiency, that is, that based on the pump work, is of course the real commercial efficiency.

For the definition of duty see Art. 363, p. 775.

Since in these engines the weight of the working fluid remains constant for the cycle, the temperatures around the cycle may be computed from $Pv = RT = 53.2\ T$, provided the temperature at one point is known.

The heat balance should show the following items:

	B.t.u.	Per cent.
(1) Total heat supplied in fuel above room temperature..	....	100
(2) The heat equivalent of the pump work....	....	....
(3) The heat equivalent of the work lost in mechanical friction....	....	....
(4) The heat carried away by the jacket water	....	....
(5) The heat lost in waste gases..	....	....
(6) The heat lost in radiation, and otherwise unaccounted for...........	....	....

For the determination of Item (1) see Chapter XIII. Item (5) in the above heat balance is computed exactly like the similar loss in the case of boilers or gas engines. See Chapters XIII and XXI.

The report should include a graphical representation of the results by means of the following curves:

	Abscissas	Ordinates.
(*a*)	Total head pumped against	Capacity in gallons or pounds per hour.
(*b*)	" "	Percentage of slip.
(*c*)	" "	Thermal efficiency.
(*d*)	" "	Duty per 1,000,000 B.t.u.

DEPARTMENT OF EXPERIMENTAL ENGINEERING, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Test of .... *Hot-air Pumping Engine.*

Date .... *Barometer* ....

Observers { ....
....
....

LOG OF TEST

Time.	Pressures.					Temperatures.			Speed.	Fuel.		Water.				
				M. E. P.			Jacket.						First Tank.		Second Tank.	
	Pressure Gauge, Lbs.	Suction Head, Ft.	Gas Pressure, Inches of Water	Working Piston.	Transfer Piston	Room	Entering	Leaving.	Revolutions per Min.	Gas-meter Reading.	Cubic Feet per Hour.	Time for Lbs.	Tare, Lbs.	Gross, Lbs.	Tare, Lbs.	Gross, Lbs.
Aver...																
Aver...																

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Test of.................... *Hot-air Pumping Engine.*

Date .... *Observers*....................

AVERAGE DATA AND GENERAL RESULTS.

Diameter of Working Piston
Area " " "
Stroke " " "
Diameter of Transfer Piston
Area " " "
Stroke " " "

Diameter of Pump Plunger
Diameter of Pump Plunger-Rod
Stroke of Pump
Scale of Indicator Springs
Thermal units per cu. ft. of fuel at standard temperature and pressure (32° F. and 14.7 Abs.) =

	Run No.					
	I	II	III.	IV.	V.	Remarks.
Suction head, feet						
Head pumped against, feet						
Total head, feet						
Water delivered, lbs. per hr., actual						
Ft.-lbs. work per hour						
Revolutions per min.						
Plunger displacement per hr., lbs.						
Slip, per cent						
Indicated H.P., working cylinder						
" " transfer "						
" " net						
Developed H.P.						
Mechanical efficiency						
Range of temp. of water-jacket ....Deg.						
Cu. ft. of fuel per hr., standard press. and temp.						
Heat supplied per hour (=100%).. B.t.u.						
Thermal equiv. ind. work......... .B.t.u						
" " " .....per cent.						
Heat absorbed by jacket-water.... B.t.u.						
" " " " . per cent.						
Radiation and loss.............. B.t.u.						
" " ....per cent.						
Heat supplied per D.H.P. per min.						
Thermal efficiency						
Duty, actual (ft.-lbs. per million B.t.u.)						

CHAPTER XXIII.

AIR-COMPRESSING MACHINERY.

402. Types of Air-Compressing Machinery. - - Air-compression or air-moving machinery may be divided into the following distinct classes: (*a*) piston air compressors, (*b*) rotary or positive blowers, (*c*) centrifugal or volume blowers or fans, (*d*) turbine compressors, (*e*) injector blowers, and (*f*) hydraulic compressors. The following paragraphs will give the main features of each one of these types. It will be found that, in general, each class is best adapted to some particular class of service. Thus, for high-pressure work the piston compressor is about the only one used; for low pressure work, generally simply the work of moving large volumes of air, the centrifugal fan is mostly employed. As far as motive power is concerned, any of the mechanisms, with the exception of classes (*e*) and (*f*), may be belt-driven or operated by any of the prime movers, as steam engines, steam turbines, electric motors, gas engines, etc. In other chapters of this book will be found full information concerning the methods of determining the power input in case any of the sources of power mentioned are used, so that this chapter will confine itself principally to the investigation of the operation of the compressing element of the machines.

403. Piston Compressors. — The construction of the compressor cylinders is very similar to that of a steam-engine cylinder. The cylinder may be single- or double-acting. Each working end is fitted with inlet (suction) valves and outlet (discharge) valves. There is a great number of different designs of cylinders, valves, and valve gears, and the student is referred to special works on the subject.* The valves may be located in the sides of the cylinder, or in the heads; in some cases the suction valves are located in the piston itself. The valves may be slide, piston, or poppet valves. In every case the aim in view is to cut down clearance as much as possible, in order to eliminate as far as possible re-expansion of the compressed air caught in the clearance spaces. (See Art. 413.)

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* See A. Von Ihering, Die Gebläse; Hiscox, Compressed Air and Its Applications; Peel, Compressed-air Plant.

There are several ways of classifying piston compressors. On the basis of the cooling system employed, we distinguish (*a*) dry compression, (*b*) compression with water injection, and (*c*) " wet" compression. Under (*a*) no cooling whatever may be employed. This method of operation is sometimes used where the final pressures attained are not high and where the resulting temperatures of compression are an advantage rather than a loss. This is the case in blowing engines for blast-furnace work. In general, however, compression under (*a*) is done with water cooling in cylinder jackets and in the intercooler, if one is used. The advantage of this method lies in the fact that the compressed air is nearly dry, which is an essential for some uses. The disadvantage is that, on account of the inefficiency of the cooling action, especially in the cylinder jackets, the work of compression is not as efficiently done as in other cases, assuming, of course, that the standard of efficiency is isothermal compression In the operation under method (*b*), a certain quantity of water in a fine spray is injected into the air for the purpose of cooling. The result is a more effective cooling than is possible under (*a*), but the moisture in the air is seriously increased, and some objection is also made on the score of increased wear of the sliding parts of the cylinder. In " wet " compression, method (*c*), the working piston does not act upon the air direct, but moves a water column by which the air is compressed. To explain the principle, in Fig. 560,* *F* is the working piston, *A* the suction valve, and *G* the discharge valve. As the piston moves to the right, the water column follows it, and air is drawn into the cylinder, together with the water seal that covers *A*, and is maintained by

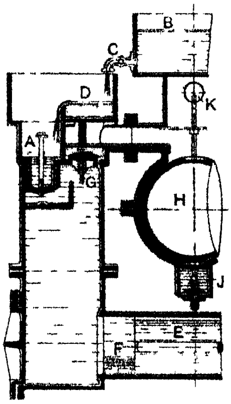

FIG 560 —"WET" AIR COMPRESSOR.

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* Von Ihering, Die Gebläse, p. 168.

the arrangement *B*, *C*, *D*. On the back stroke, *A* closes, the water column compresses the air into the vertical chamber at the end of the cylinder until *G* opens, after which the piston forces all of the air, together with a small quantity of water, into the discharge space. *K* is the high-pressure pipe, while *H* is a float arrangement to allow the water pumped over to drain out. The advantages of this system— that is, lower temperature of compressed air, higher volumetric efficiency —are fully balanced by slower speed, greater water and power consumption, etc.

The efficiency of cooling has great influence upon the power input for a given amount of compressor work to be done. This fact has led to the subdivision of the total compressor work among several cylinders where the air is to be highly compressed. Depending upon the number of cylinders used for the work, we then distinguish single-stage, two-stage, or multistage compressors. A common pressure to which air is compressed for a variety of uses is about 125 pounds. In this case, two- or perhaps three-stage compression should be employed. The former would be more common, the air being compressed up to about 40 pounds in the first stage and to 125 pounds in the last. Unless the pressures are very high, as in liquid-air work, it is not common to exceed two stages because the gains due to efficient cooling are, beyond the second or third stage, soon overbalanced by mechanical losses. The several cylinders may be arranged one behind the other, in which case we have what is known as a tandem arrangement, or they may be placed side by side with the intercooler between them, in which case we have the duplex arrangement. A two-stage compressor of the former design, steam-driven, is shown in Fig. 561, where *D* is the steam cylinder, *A* the low-pressure compression cylinder, *B* the intercooler, and *C* the high-pressure compressor cylinder. When the engine end is also compounded, a favorite arrangement is to cross-compound the engine and to place an air cylinder behind each steam cylinder in tandem. This makes a very good design as far as stress conditions in frames and crank shafts are concerned.*

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* The question of the best arrangement of cylinders in motor-, belt-, or steam-driven air compressors is discussed by E. W. Koester in *Zeitschrift des Vereins deutscher Ingenieure*, Jan. 23, 1904.

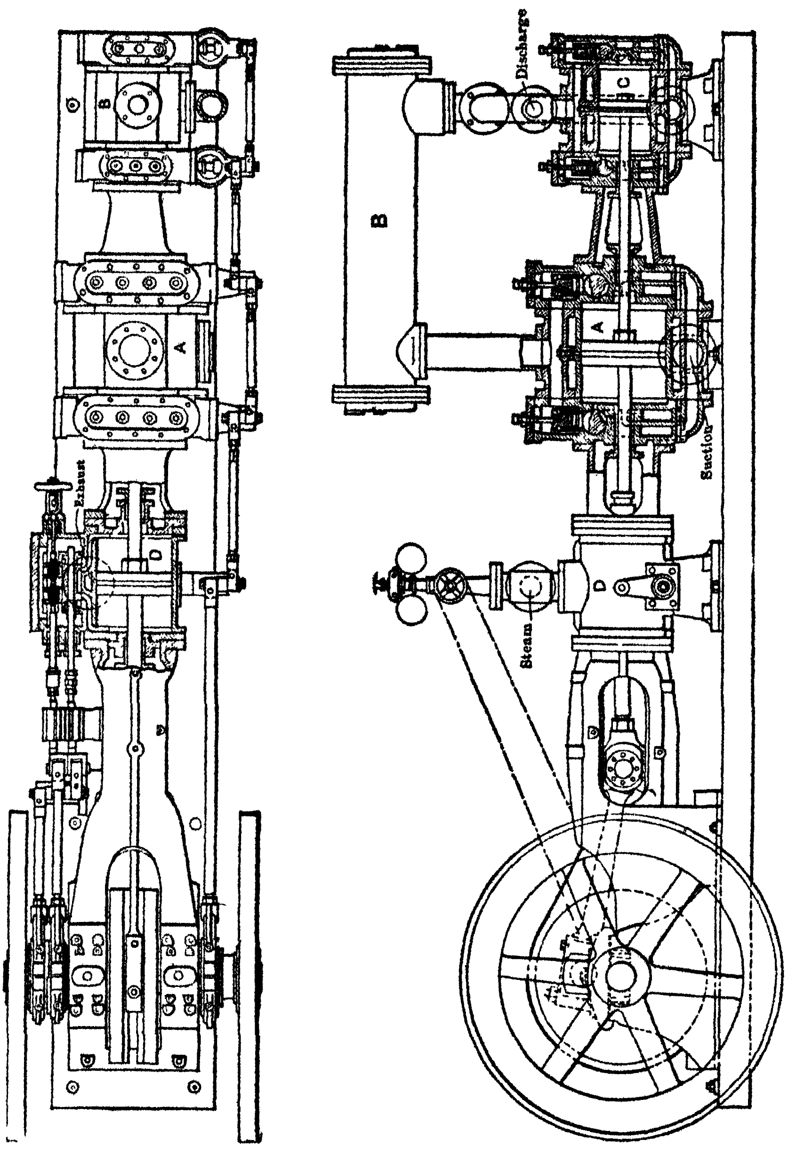

FIG. 561.— TWO-STAGE TANDEM AIR COMPRESSOR.

A special form of the piston compressor is the *air pump* used in condenser work. The conditions of operation are somewhat different, in so far as in the ordinary compressor the suction pressure is constant, while the discharge pressure may vary; in the air pump the suction (condenser) pressure varies while the pump delivers against a constant pressure,— that of the atmosphere. Otherwise, the conditions of operation are similar.

404. **Rotary or Positive Blowers.** — These generally consist of two blades, pistons, or displacers, *A* and *B*, which revolve in opposite directions, as indicated in Fig. 562. They are actuated by gearing or other devices outside of the case *C*. Each displacer on the one side maintains continuous contact with the inner surface of the case and on the other side with the mating displacer. As a consequence, a certain volume of air is picked up on the suction side (the volume represented by the space *E* in Fig. 562) and transferred to the discharge side *F*. The pressure that can be maintained at *F* depends, of course, upon the amount of compressed air used and upon the speed of rotation of the displacers. If the blower delivers into a closed system, the pressure will build up until the leakage back past the contact surfaces of the displacers prevents further increase. Hence a close fit at the contact surfaces is essential for the attainment of highest pressure. The gain thus made, may, however, be counteracted by an increased friction loss, decreasing the total efficiency of operation. These machines are, therefore, not adapted to high-pressure work, and 10 pounds per square inch is about the outside limit. For lower pressures than this, but higher than those that can be efficiently handled by the centrifugal blower, they are quite satisfactory on account of their large volume capacity, in spite of the fact that the volumetric efficiency may not be over 75 per cent and is frequently lower than that.

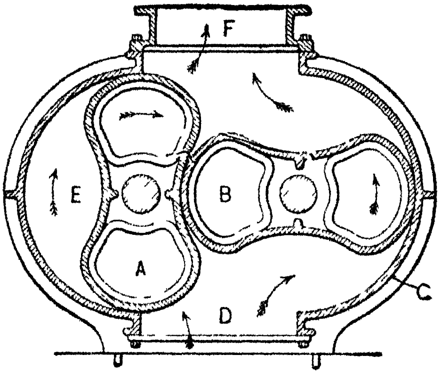

FIG. 562. — ROTARY OR POSITIVE BLOWER.

405. Centrifugal or Volume Blowers. — Two fundamental types of these blowers are recognized: (*a*) the disk or propeller fan, in which the air moves through the fan practically parallel to the shaft and perpendicular to the plane of rotation of the wheel; and (*b*) the machine generally known simply as the blower or fan, in which the air enters the fan wheel at right angles near the center, is deflected through a right angle as it goes through the wheel and leaves the periphery of the wheel at high speed owing to the centrifugal force imparted.

Disk fans are constructed either with straight or curved blades, as shown in Fig. 563. The action of these fans is the reverse of that of a windmill. In the latter, the wind moving with a certain velocity imparts to the wheel a certain rotative speed which is utilized to do certain work. In the disk fan, a certain amount of work is put in at the shaft, the wheel attains a certain rotative speed, which produces a certain velocity of translation in the air in which it moves. These machines, therefore, produce a pressure difference which must be proportional to the velocity of the moving air. They are, however, not adapted to give large pressure difference, and are consequently mostly used for ventilating purposes, where the pressure or depression produced need not exceed .2 to .5 inches of water. As compared with the ordinary centrifugal blower, their power consumption for a given quantity of air moved under given conditions is less.

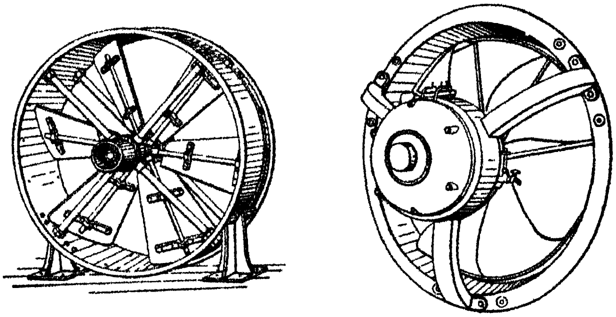
FIG. 563. — TYPES OF DISK OR PROPELLER FAN.

There are a number of different types of fans or blowers. They may be divided into two classes, — those with and those without *diffusers.* Under each class there are then two subclasses, based upon the blade form, whether straight or curved. A fan is said to be without a diffuser if the fan casing is concentric with the wheel all around, except, of course, at the point of outlet. When, on the other hand, the casing forms a gradually increasing scroll, so that the space between periphery of wheel and of casing gradually increases toward the outlet, the fan is said to have a diffuser. The purpose of this construction is to gradually decrease the velocity of the air leaving the fan wheel itself and to convert some of the velocity head into pressure head. Hence the use of this construction in all pressure blowers. The advantage of the curved blade lies mainly in the avoidance of shock as the air enters the wheel. This not only increases the efficiency but makes the operation smoother and less noisy. Figs. 564 to 566 show several examples of the constructions mentioned above.

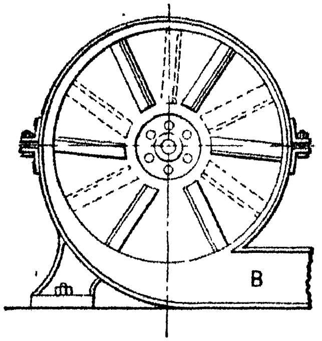

Fig. 564. — Fan with Diffuser, Straight Blades.

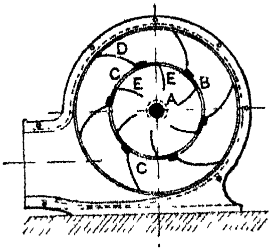

Fig. 565. — Fan without Diffuser, Curved Blades.

Any fan may serve to create a pressure at the outlet, or to produce a depression (pressure below atmosphere) at the inlet. In the former case it is called a blowing fan or blower, in the latter case an exhaust fan or exhauster.

With reference to the pressures or depressions produced, centrifugal fans are sometimes classified as low-pressure or volume fans and as high-pressure fans. The distinction is, however, not clearly drawn, since the pressure may, in most cases, be considerably changed by changing the speed of rotation. In actual service the pressures or depressions used range about as follows: .5 inches or less for the ventilation of schools, theaters, factories, etc.; 3-6 inches for mine fans; 4 8 inches for forge fans; and 8-15 inches for fans used for melting-furnaces, like cupolas, etc.

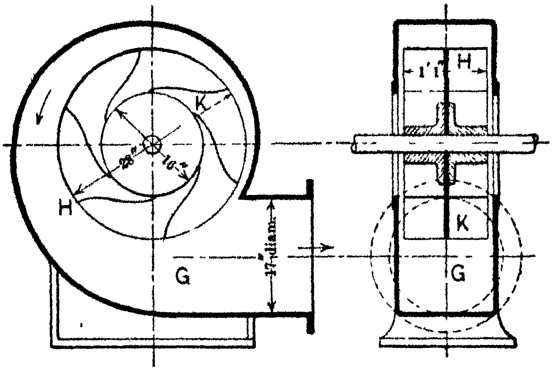

FIG. 506.—FAN WITH DIFFUSER, CURVED BLADES.

406. Turbine Compressors.—This type of compressor has been developed in the last few years. The principle of its operation is practically the same as that of the multistage turbine pump (see Chap. XXV). There is, however, this important difference: In the water pump there is no change in the density of the medium, and each wheel causes the same absolute pressure increase in the water. In the case of the turbine compressor, on the other hand, the density of the medium changes from stage to stage, and, assuming that the temperature does not increase, each wheel only causes the same relative pressure increase. Thus, if each wheel causes a relative pressure increase of 10 per cent, it will take 10 stages to reach an absolute pressure of 2.6 atmospheres, and 10 more stages to reach a final pressure of 7 atmospheres absolute. In practice, on account of temperature increase and other losses, from 25 to 30 wheels or stages are required to attain the latter pressure. Where the final pressures are quite high, the wheels are usually made in groups of different diameters, the largest wheels forming the first three or four stages. The peripheral speed of such wheels may be up to 460 feet per second, while the wheels in the final stages have a rim speed of about 330 feet per second. The rotative speeds for high-pressure work may be from 3000 to

4000 r.p.m. For low degrees of compression, all the wheels are made of the same diameter; thus the Rateau turbine blower shown in Fig. 567 consists of four stages and compresses the air to about 10 pounds per square inch above the atmosphere when making 1875 r.p.m. A favorite method of driving such a compressor is by direct connection to a steam turbine, as illustrated in Fig. 567.

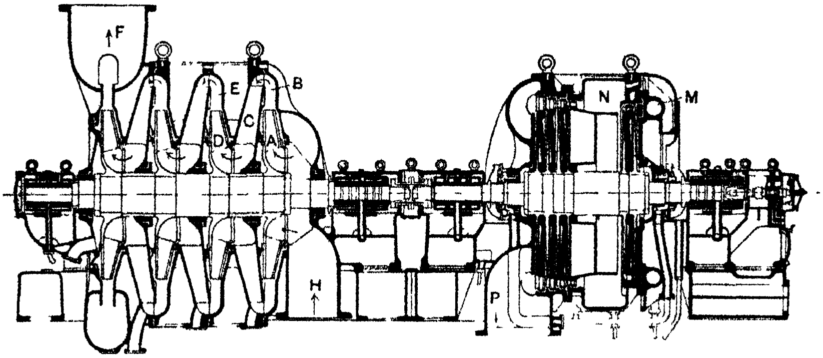

Fig. 567.—Rateau Turbine Compressor.

The turbine blower has an advantage over the piston compressor in so far as it is possible to cool more thoroughly, since there are usually many stages and the temperature increase in each stage is small. On the other hand, the turbine compressor, analogous to the steam turbine, shows greater friction losses than the piston compressor. These may be so great in a compressor not cooled as to cause final temperatures in the compressed air even above those due to adiabatic compression. Further, a comparison of turbine and piston compressors should also take into account the characteristics of the machines used to drive them, which in the former case is usually a steam turbine, in the latter case a reciprocating engine. As a final result it may be said that, at the present state of the development of the turbine compressor, the piston compressor with good compound engines, running condensing, has slightly the better of the argument on the score of efficiency.*

407. Injector Blowers. — The principle of action of these devices is exactly the same as that of the injector used for boiler feeding.

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* There is an interesting report upon the relative efficiency of these two compressor types in the *Zeitschrift d. V. d. I.* for Oct. 30, 1909.

The working medium — steam, compressed air, or water — must be under considerable pressure. This pressure is converted into velocity by passing the medium through a nozzle. The high-velocity jet carries along with it the air immediately surrounding it, and, if the space is confined, will create a vacuum in the latter. If this space is then connected by a suction pipe to some other space, the air will be removed from the latter, and the apparatus acts as an exhauster. If, on the other hand, the space around the nozzle or nozzles is unconfined, but the outlet tube of the device is connected to some restricted space, the air pressure in the latter will be increased and the apparatus acts as a blower.

Among the advantages possessed by these devices may be named the following: Elimination of prime mover, small cost of repair, small space required for installation, easy regulation of quantity of air handled, etc Injector blowers are made in capacities ranging from 0.2 to 600 cu. ft. of air per minute, and, according to Ihering, require a steam pressure of from 75–90 lbs., or a water pressure of from 45–75 lbs., or an air pressure of from 45–60 lbs. per square inch.

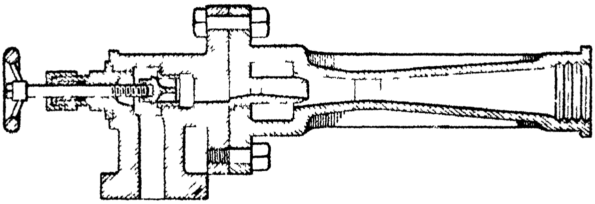

FIG. 568.—INJECTOR BLOWER

The construction of an injector blower, used for the production of forced draft under boilers, is shown in Fig. 568. Fig. 364, p. 516, shows one construction used for exhausters.

408. Hydraulic Compressors.*—The method of operation of these compressors is best illustrated by means of Fig. 569. The installation consists first of a receiving bay or chamber in which is placed the suction head. The latter may be constructed in several ways.

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* See Peele, "Compressed Air Plant," Chap. XV, for a discussion of the construction and efficiency of operation of several installations of this type of compressor in this country.

In this case, it consists of a ring pipe *a*, to the inner circumference of which are fastened a large number of horizontal tubes *c*. As the water enters the vertical pipe leading into the shaft, suction is created at the opening of the tubes *c* and the air will be drawn into *a* and *c* through *b*. This air becomes mixed with the falling water in small bubbles and is compressed in the descent, according to the height of the fall. The separation tank is placed at the bottom of the shaft. In it the direction of the falling water is, by means of baffle plates, changed to the horizontal, and the air separates from the now comparatively quiet mass of water. The latter finds its way out through the open bottom of the separator tank and rises in the shaft to the level of the tail race. The compressed air gathers

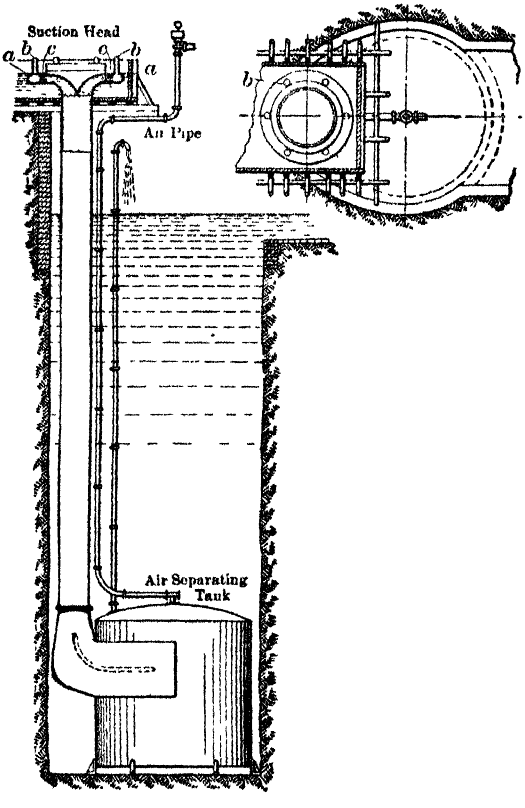

FIG. 569.—HYDRAULIC AIR COMPRESSOR.

in the upper part of the tank and is taken off as desired by means of the air-pipe.

One great advantage of this type of compressor is found in the elimination of the prime mover. The compression pressure attained depends directly upon the distance from the disengagement surface in the separator tank to the level of the tail race, and is, therefore, independent of the head of water from the receiving bay. The air is compressed isothermally, since each air bubble is surrounded by the water; and, in spite of the fact that at some parts of the compressing operation the air is probably saturated with moisture, it is found that compressed air delivered by these compressors is comparatively dry, which is probably due to the fact that at the low disengagement temperature the absolute moisture content cannot be high.

These installations have proved very efficient, over 90 per cent having been shown in case of high heads.

409. Physical Characteristics of Air. — It has already been pointed out (Chap. XI, under Laws of Gases) that air may be considered as a permanent gas following the law

$$\frac{Pv}{T} = \text{constant} = R. \tag{1}$$

In this equation, substitute for standard conditions, $P = 144 \times 14.7 = 2116.8$ lbs. per square foot, $T = 460 + 32 = 492°$ F., and specific volume $v = \frac{1}{.0807}$, where .0807 is the weight of a cubic foot of *dry* air under standard conditions. Then

$$\frac{Pv}{T} = \frac{2116.8}{492 \times .0807} = R = 53.35.^* \tag{2}$$

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* The value of R changes slowly both with pressure and temperature. Another expression for R is $A \cdot R = C_p - C_v = C_v\left(\frac{C_p}{C_v} - 1\right) = C_v(\gamma - 1)$. The values of γ and C_v change with temperature See Chap. XXI under Specific Heats. The change with pressure is less important, although appreciable at very high pressures. Ihering gives the following figures, computed from an equation given by Antoine in Comptes Rendues, 1890:

p =	1 atm.	40 atm.	60 atm.	80 atm.	100 atm.	200 atm.
R =	53.30	53.30	54.25	55.30	56.40	62.30

These notes refer, of course, only to the case of dry air.

From this
$$v = \frac{53.35\,T}{P}\text{ cu. ft. per lb.,} \tag{3}$$

and weight per cubic foot of *dry* air

$$\delta = \frac{1}{v} = \frac{P}{53.35\,T} = .01874\,\frac{P}{T}\text{ lbs.} \tag{4}$$

Eq. (4) enables us to compute the weight per cubic foot of *dry* air for any condition of P and T.

Example. — Pressure = 80 lbs. per square inch by gauge, barometer = 28.5 inches Hg, temperature = 120° F.

$$P = \left(80 + \frac{28.5}{2.0378}\right)144 = 13{,}577\text{ lbs. per square foot; } T = 460 + 120 = 580°.$$

$$\text{Weight per cubic foot} = \frac{13{,}577}{53.35 \times 580} = .438\text{ lbs.}$$

Under nearly all conditions of practical operation, however, air is not dry but has associated with it a certain amount of moisture (water vapor, humidity). This fact not only changes the weight per cubic foot of the mixture for given P and T conditions from the value computed for dry air, by Eq. (4), but it also affects the relations involved in the compression and expansion of air.

For any given values of P and T, a given space can contain, as a maximum, a certain weight of water vapor. When the maximum weight is present, the vapor is said to be saturated. In actual practice, however, it is found that this 100 per cent degree of saturation is rarely ever attained, but that the weight of water vapor is less than the maximum. The vapor is then said to be only partly saturated (it is, as a matter of fact, superheated), and the degree of saturation is expressed by the ratio of the weight of water vapor actually contained in a given space to the maximum weight that the space can contain under the P and T conditions existing.*

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* It is common usage to say that the air in any given space has a certain capacity for water vapor. As a matter of fact, the presence of air in any given space has nothing to do with the amount of water vapor the space contains; that is, the weight of moisture in the space is controlled only by the pressure and temperature conditions existing in the space. It is more correct to say, therefore, that the *space* is either completely or partly saturated. In the first case, it contains as much water vapor as can exist in the saturated state for the existing temperature. In the latter case, the weight of water vapor present is less than this, and it is consequently superheated, since it is at a higher temperature than that of saturation for the pressure existing.

Humidity may be expressed in two ways, — *absolute* and *relative* humidity. Absolute humidity is the weight of water vapor that any given space (one cubic foot usually) actually contains. If the vapor is saturated, the absolute humidity may be found from the Steam Table. Thus in Table 3, I, Appendix, if the temperature of the air is 75° F., the density of the water vapor is seen to be .001346 lbs. per cubic foot; that is, the absolute humidity for 100 per cent saturation is .001346. Relative humidity is the ratio between the actual weight of water vapor that a given space contains under given conditions and the maximum weight that the same space may contain under the same conditions. Thus, if the weight actually present at a temperature of 75° is only .000587 lbs. per cubic foot (absolute humidity = .000587), the relative humidity

$$= \frac{.000587}{.001346} = .436 = 43.6 \text{ per cent.}$$

410. Determination of Humidity. — Absolute humidity may be found by drawing a definite volume of air through tubes containing chloride of calcium or sulphuric acid. The increase in weight of the absorbent determines the actual weight of water vapor in the quantity of air used. The method, while accurate, is not of easy application in every-day practice and little used.

It is easier to determine relative humidity, for which there are several methods. The general method is based upon the assumption that the vapor pressures of water at any one temperature are directly proportional to the absolute weights of water vapor present in a given space. This assumption is very nearly true. If we let w_s = the maximum possible weight of water that the space can contain (that is, at 100 per cent saturation), at a given temperature, w = the weight actually contained, p_s = the vapor pressure for 100 per cent saturation, and p the actual vapor pressure, then

$$\frac{p}{p_s} = \frac{w}{w_s}, \tag{5}$$

and the relative humidity $= 100\frac{p}{p_s}$ per cent.

p_s may be obtained from steam tables as soon as temperature is known, but p must be found by experiment. One method of doing this is to find the *dew point* by cooling the air down to saturation. A very simple instrument for this purpose is illustrated by Gramberg,* Fig. 570. A small vessel A, containing ether, has fastened

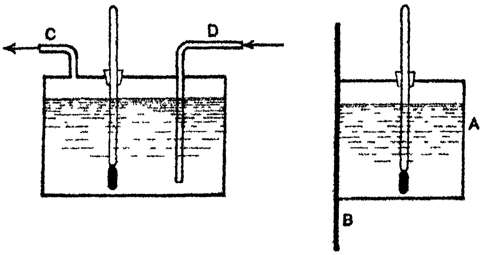

FIG. 570.—DEW POINT APPARATUS.

to it a nickel mirror B. By means of an aspirator bulb connected at C, air is drawn through the apparatus, entering at D. The vaporization of the ether causes cooling, which chills the mirror and the air surrounding it. At the instant that the air in contact with the mirror reaches 100 per cent saturation, fog will appear on the mirror, and the temperature t of the ether is noted. For this temperature t the maximum vapor pressure may be obtained from steam tables, which determines the factor p in the above equation. Thus, suppose that the dew point is found to be 55°, with the air temperature at 75°. From the steam table, p_s = .4288 lbs. per square inch, and p (= p_s for 55°) = .2170 lbs. per square inch. Then relative humidity $= \frac{.2140}{.4288} = .50 = 50$ per cent.

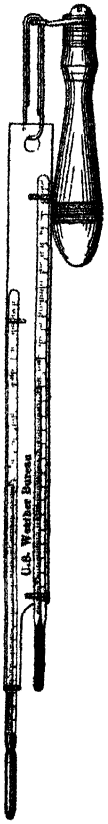

FIG. 571.—PSYCHROMETER (WET AND DRY BULB THERMOMETER).

A second method of determining relative humidity, that by *wet and dry bulb thermometer*, is radically different. The apparatus is called a *psychrometer* and is more used than the dew-point apparatus. It consists essentially of two thermometers fastened to a frame (see Fig. 571), which represents the swing type used by the U. S. Weather Bureau. The bulb of one of the thermometers is surrounded by a piece of thin muslin, which, during the use of the instrument, is kept thoroughly wetted with water. The bulb of the other thermometer is left free. If the water vapor in the air is saturated, the two thermometers will continue to indicate the same temperature, which is that of the surrounding air. But if the relative humidity is less than 100 per cent, vaporization of water takes place from around the wet-bulb. This action is necessarily combined with the rendering latent of a certain amount of heat, which causes the wet-bulb temperature to sink. The lower this temperature, the greater the absorption of heat from the surrounding bodies, and at some point there must be established an equilibrium between the heat which the wet bulb is losing and that which it is gaining by conduction. This equilibrium temperature is called the wet-bulb temperature for the conditions existing, and *must not be confused with the dew point* for the same conditions. From the difference between the dry-bulb temperature t and the wet-bulb temperature t', it is possible to compute directly the relative humidity for the conditions existing from the equation

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* Gramberg, Heizung & Lüftung von Gebäuden.

$$e = \left[p_{t'} - .000367\, B\,(t - t')\left(1 + \frac{t' - 32}{1571}\right)\right] \div p_t \qquad (6)$$

given in the Psychrometric Tables published by the U. S. Department of Agriculture, 1900. Here

e = relative humidity;
p_t = saturation pressure at temperature t, inches Hg;
$p_{t'}$ = the saturation pressure at the temperature t', inches Hg (taken from a steam table);
B = barometer pressure, inches Hg;
t = air temperature = temperature of dry bulb;
t' = wet-bulb temperature.

Example.—Let $t = 75°$ F., $t' = 55°$, $B = 30''$ Hg. From the Steam Table, Appendix, for 55°, $p_t = .873$, $p_{t'} = .4357$; then

$$e = \left[.4357 - .000367 \times 30 \times 20\left(1 + \frac{23}{1571}\right)\right] \div .873 = .244 = 24.4 \text{ per cent.}$$

At a temperature of 75° F., the weight per cubic foot of saturated aqueous vapor is .00134 lbs., hence the absolute humidity under the conditions stated is

$$.00134 \times .244 = .00033 \text{ lbs.}$$

In practice, instead of making the above computation, humidity tables are used. There are a number of these, computed by different authorities, none of which exactly agree, although the difference is negligible for engineering work. Table 5, in the Appendix, is taken from the above-mentioned Psychrometric Tables of the Department of Agriculture. It is extensive enough for ordinary work. The relative humidity percentages even in this table do not exactly agree with results obtained by the formula above. Thus for 75 degrees and a depression of 20 degrees, the table gives 24 per cent, while the formula gives 24.4 per cent. Some of this difference may be due to barometric corrections which were not applied in the computation above. In connection with the formula and the table, it should be stated that the results are strictly applicable only to readings obtained with the same form of psychrometer, that is, the swing type. To use this instrument, thoroughly wet the muslin around the wet bulb. Then by means of the handle at the top, which can be extended at right angles, whirl the frame with the thermometers rapidly for 15 or 20 seconds. Stop and quickly read the wet bulb. Repeat until at least two consecutive readings agree very nearly.

Fig. 572 gives a graphical representation of a humidity table, constructed for 30-inch barometer, by means of which quick determinations, intermediate between those of the table in the Appendix, may be made. As an example of its use, if the air temperature = 80° (point *A*) and the wet-bulb temperature = 60° (point *B*), the relative humidity will be 29 per cent, while the absolute humidity, the weight of water vapor per cubic foot, is $\frac{4.5}{10,000} = .00045$ pounds (point *D*). On the consideration that this weight represents the maximum, that is, 100 per cent saturation (point *C*), we may also locate the dew point *F*, at slightly over 42 degrees.

411. The Theoretical Work of Compressing Air.* — The formulæ developed in this article apply only to the theoretical work of compressing air and enable the student to compute theoretical horse power necessary. No account is here taken of the effect of clearance, nor is the effect of the heating of the intake air upon the

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* The text of this article is largely adapted from Ihering, "Die Gebläse."

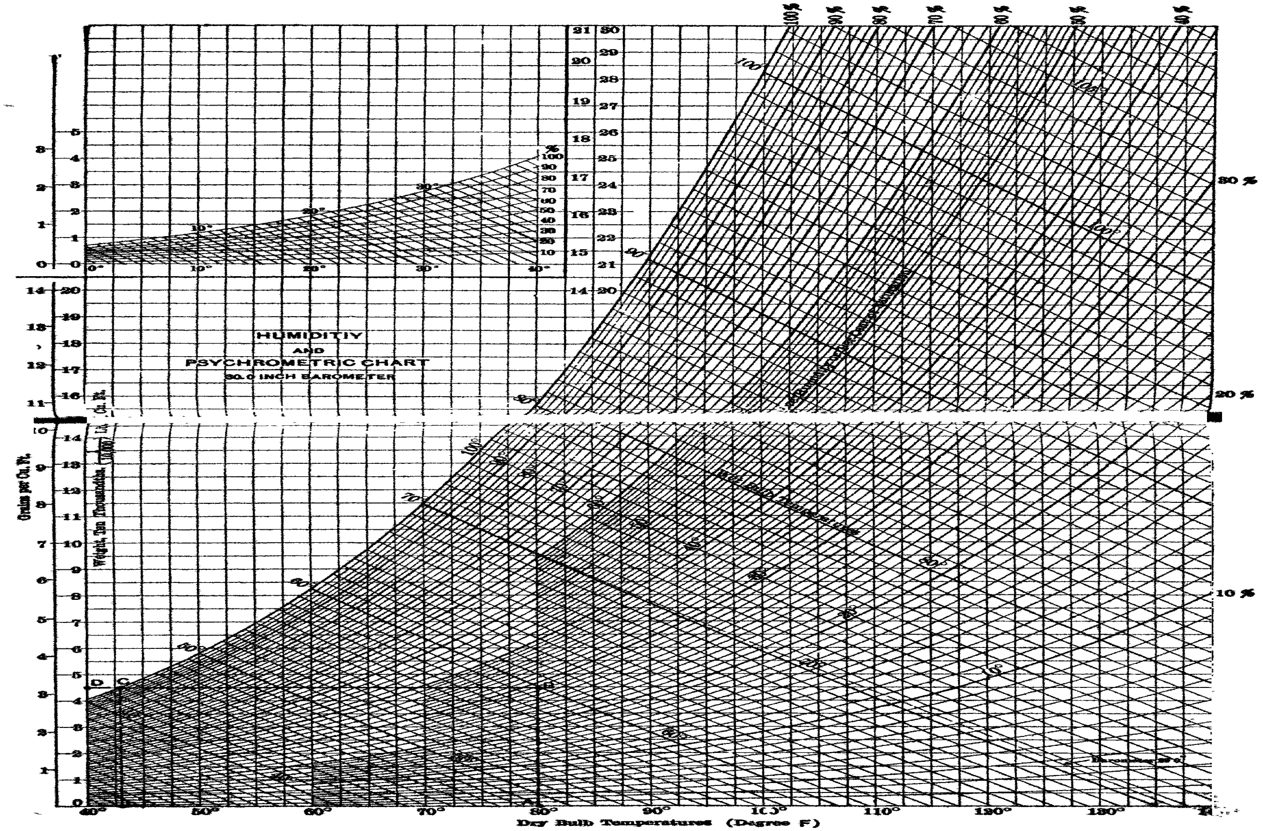
HUMIDITIY
AND
PSYCHROMETRIC CHART
30.0 INCH BAROMETER
Grains per Cu. Ft.
Weight, Ten Thousandths ($\frac{1}{10,000}$) Lb. Cu. Ft.
Dry Bulb Temperatures (Degree F)
40°
50°
60°
70°
80°
90°
100°
110°
120°
130°
0°
10°
20°
30°
40°
100 %
90 %
80 %
70 %
60 %
50 %
40 %
30 %
20 %
10 %
D
C
A

FIG. 572.

volumetric efficiency considered. In other words, it is assumed that there is no clearance and that the volumetric efficiency is 100 per cent. The changes produced in the results obtained when both of these factors are considered are outlined in Art. 413. It is hardly necessary to add that friction losses in the machine are also neglected.

The work of compression consists of three parts: W_1, the work of charging the cylinder; W_2, the work of raising the pressure of the air from some pressure p_1 and volume V_1 to some pressure p_2 and volume V_2; and W_3, the work of forcing the compressed air out of the cylinder. W_1 is done on the side of the piston opposite to that on which W_2 and W_3 are performed in the same cycle, hence the total work of compression

$$W = W_2 + W_3 - W_1.$$

Let P_1 = absolute pressure during suction stroke, pounds per square foot.
P_2 = absolute pressure during discharge stroke, pounds per square foot.
V_1 = volume of one pound of air at pressure P_1, end of suction stroke, in cubic feet.
V_2 = volume of one pound of air at pressure P_2, end of compression, in cubic feet.
T_1 = absolute temperature at end of suction stroke.
T_2 = absolute temperature at end of compression.
S = stroke of piston, in feet.
S_1 = stroke during which air is discharged, in feet.
F = area of piston, in square feet.
$r = \frac{V_1}{V_2}$ = ratio of compression.

A. Compression of Dry Air. — *(a) Isothermal compression. Work per pound of air.*

$$W_1 = FP_1S = P_1V_1 \text{ ft.-lbs.} \tag{7}$$

$$W_3 = FP_2S_1 = P_2V_2 \text{ ft.-lbs.} \tag{8}$$

But $P_1V_1 = P_2V_2$, hence $W_1 = W_3$.

From Art. 177, p. 330,

$$W_2 = W = P_1V_1 \log_e \frac{V_1}{V_2} = P_1V_1 \log_e \frac{P_2}{P_1} \qquad (9)$$

$$= RT_1 \log_e \frac{P_2}{P_1} = RT_1 \log_e r \text{ ft.-lbs.} \qquad (9a)$$

The term " free air " refers to the volume of air under suction conditions of pressure and temperature.

To compute the work W_I per *cubic foot of free air*, divide Eq. (9) by the specific volume of air for P_1 and T_1, that is, by V_1. Then

$$W_I = P_1 \log_e \frac{P_2}{P_1}. \qquad (10)$$

To find the work W_{II} done per *cubic foot of compressed air*, we have

$$W_{II} = P_1V_1 \log_e \frac{P_2}{P_1} = P_2V_2 \log_e \frac{P_2}{P_1}.$$

Since V_2 = volume of compressed air, it follows that

$$W_{II} = P_2 \log_e \frac{P_2}{P_1}. \qquad (11)$$

(*b*) *Adiabatic compression. Work per pound of air.*

In this case:

$$W_1 = P_1V_1 \text{ ft.-lbs.} \qquad (12)$$

$$W_3 = P_2V_2 = P_1V_1\frac{T_2}{T_1} = P_1V_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \text{ ft.-lbs.} \qquad (13)$$

From Art. 177, p. 331, the work of compression is

$$W_2 = -\frac{P_1V_1}{\gamma - 1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}\right] = \frac{P_1V_1}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]. \qquad (14)$$

The total work, then, is

$$W = \frac{P_1V_1}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + P_1V_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - P_1V_1$$

$$= P_1V_1\frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = \gamma W_2 \text{ ft.-lbs.} \qquad (15)$$

Work W_I done per *cubic foot of free air,*

$$W_I = P_1 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \text{ft.-lbs.} \tag{16}$$

Work W_{II} done per *cubic foot of compressed air at temperature* T_2,

$$W_{II} = \frac{P_1V_1}{V_2}\,\frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \text{ft.-lbs.}$$

$$= P_1 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right) - \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}}\right] \text{ft.-lbs.} \tag{17}$$

The compressed air may not remain at the temperature T_2, but may be cooled back to some temperature T_2'. The volume will then change from V_2 to some value V_2', so that

$$\frac{V_2}{V_2'} = \frac{T_2}{T_2'}, \quad \text{from which} \quad V_2' = V_2\frac{T_2'}{T_2}. \tag{18}$$

Work W_{III} per *cubic foot of compressed air cooled back to some temperature* T_2', then, is

$$W_{III} = \frac{P_1V_1}{V_2'}\,\frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = \frac{P_1V_1}{V_2}\,\frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]\frac{T_2}{T_2'}$$

$$= P_1 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right) - \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}}\right]\frac{T_2}{T_2'} \text{ft.-lbs.} \tag{19}$$

For the particular case that the compressed air is cooled back to T_1, substitute $T_2' = T_1$ in the second form of Eq. (19) above.

The work W_{IV} done per *cubic foot of compressed air cooled back to* T_1 will then be found to be

$$W_{IV} = P_2 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \text{ft.-lbs.} \tag{20}$$

Comparing this with Eq. (16), it will be seen that

$$\frac{W_I}{W_{IV}} = \frac{P_1}{P_2}, \quad \text{or} \quad W_{IV} = W_I\frac{P_2}{P_1} \text{ft.-lbs.} \tag{21}$$

The *temperature rise during compression* may be found from

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}, \tag{22}$$

which is derived from a combination of the equations $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$ and $P_1V_1^\gamma = P_2V_2^\gamma$.

The examples at the end of Art. 412 will show that more work is required, for the same amount of free air compressed, when the compression line follows the adiabatic than when it follows the isothermal. This becomes apparent also when we consider the compressor-indicator diagram. In Fig. 573, BC is the isothermal and BC' the adiabatic compression line. The work of compression is in the first instance represented by the area $ABCD$, in the second by the area $ABC'D$. Since in most applications of the use of compressed air the latter cools down practically to the temperature at B before use, it follows that the area BCC' is lost work. The aim in actual practice should, therefore, be to compress as nearly along an isothermal as possible; hence the use of cooling during compression. The exponent n, in the equation $pv^n =$ const., is equal to 1.0 for the line BC, and to 1.408 for the line BC'. In any given case the value of n for the actual compression line BC'' will be between these values. The more perfect the cooling, the nearer will the ideal value $n = 1.0$ be approached.

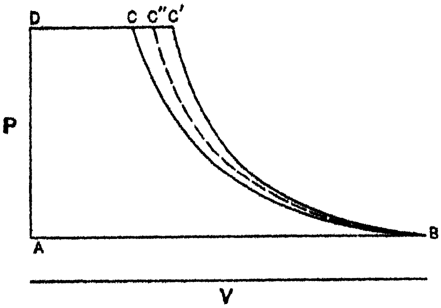

FIG. 573.

B. Compression of Moist Air. — The presence of moisture in air modifies the work formula above developed. The differences introduced are, however, so small in the average case that they may be neglected even for high-pressure work. The example at the end of Art. 412 shows this clearly. For the purpose, however, of giving the method of computation involved in the case, the following formulas were adapted.

Let P = absolute pressure per square foot of any given volume of moist air, P_a the absolute partial pressure per square foot due to the dry air contained in the volume, and P_w the absolute partial pressure due to the water vapor. Then

$$P = P_a + P_w. \tag{23}$$

P_w must in every case be determined by one of the methods given in Art. 410, after which P_a becomes known.

The weight of a cubic foot of saturated moist air may now be computed as follows:

Let δ = weight of a cubic foot of dry air, in pounds at a temperature T;

δ_1 = weight of a cubic foot of moist air, in pounds at a temperature T; and

δ_2 = weight of a cubic foot of the water vapor contained in the saturated moist air. This is found from steam tables.

From Art. 409, Eq. (4),

$$\delta = .01874\frac{P_a}{T}. \tag{24}$$

Hence

$$\delta_1 = \delta + \delta_2 = .01874\frac{P_a}{T} + \delta_2$$

$$= .01874\frac{P - P_w}{T} + \delta_2 \text{ lbs. per cu. ft.} \tag{25}$$

If the air is not saturated, but the relative humidity = e, then

$$\delta_1 = .01874\frac{P - eP_w}{T} + e\delta_2. \tag{26}$$

Example. — Absolute pressure P = 14.7 lbs. per square inch = 2117 lbs. per square foot; t = 100° F., so that $T = 460 + 100 = 560°$. Find δ_1 for the fully saturated condition, and for relative humidity = 50 per cent.

For 100° F., δ_2 = .002851 lbs. per cubic foot, while for saturation $P_w = 144 \times .946 = 136.2$ lbs. per square foot, both figures derived from the steam table.

For the saturated condition, then,

$$\delta_1 = .01874\,\frac{2117 - 136.2}{560} + .002851 = .0692 \text{ lbs. per cu. ft.}$$

For the relative humidity of 50 per cent, $e = .5$,

$$\delta_1 = .01874\,\frac{2117 - .5 \times 136.2}{560} + .5 \times .002851 = .0699 \text{ lbs. per cu. ft.}$$

The weight of absolutely dry air would have been

$$\delta_1 = .01874\,\frac{2117}{560} = .0709 \text{ lbs. per cu. ft.}$$

The work of compression of moist air is composed of two parts: (*a*) the work required to compress the *dry* air contained in a given volume of moist air, and (*b*) the work required to compress the water vapor contained in this volume. For the work under (*a*) the formulas above developed for dry air directly apply. Work under (*b*) may be found as follows: The water vapor before compression has the partial pressure P_w, properly computed for the relative humidity existing, and the temperature T_1. At the end of compression its volume is V_2, and it must have the temperature T_2, the same as the air. With this data we may compute the partial pressure P_w' at the end of compression from

$$\frac{T_2}{T_1} = \left(\frac{P_w'}{P_w}\right)^{\frac{\gamma-1}{\gamma}} \qquad (27)$$

In this case the value of $\gamma = \frac{C_p}{C_v}$ for superheated steam in the range T_1 to T_2 (see the article on specific heat, in Chap. XXI).

From Eq. (15), the work of compressing *one pound of dry air* would be

$$W_a = P_a V_1 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_a'}{P_a}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \text{ft.-lbs,} \qquad (28)$$

in which P_a' is determined by an equation of the form of Eq. (27).

For superheated water vapor, the work of compressing one pound will by analogy be

$$W_b = P_w V_1 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_w'}{P_w}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \text{ft.-lbs.} \qquad (29)$$

If now in one pound of moist air there are G_1 pounds of dry air and G_2 pounds of water vapor (so that $G_1 + G_2 = 1.0$), the *work of compressing one pound of the moist air* will be

$$W = G_1 W_a + G_2 W_b \text{ ft.-lbs.} \tag{30}$$

The above method is probably not quite correct on account of heat interchanges between air and water vapor, the values of γ being different. A more exact and simpler way is to compute an average value of γ for the mixture of air and water vapor, but this does not separate the two amounts of work to bring out their relative importance. The error in the first method is in any case not great, and it may be used for the purpose of obtaining an idea of the relative proportion of the two amounts of work.

$$\text{Average } \gamma = \frac{C_p' + mC_p''}{C_v' + mC_v''}, \tag{31}$$

in which C_p' and C_v' are the specific heats for air, and C_p'' and C_v'' those for the superheated water vapor, while m is the weight ratio of water vapor to air in one pound of moist air $= \frac{G_2}{G_1}$. This ratio may be computed from the equation

$$m = \frac{R_1}{R_2} \cdot \frac{P_w}{P_a} = \frac{53.35}{85.86} \frac{P_w}{P_a} = .621 \frac{P_w}{P_a} \tag{32}$$

The theoretical work required for one pound of the moist air between total pressures P_1 and P_2 is then computed from Eq. (15), except that for the value of γ for dry air the average value of γ for the mixture is substituted.

412. Theoretical Compressor Horse Power. — Let the theoretical work required to compress one cubic foot of free air under given condition be in general W foot-pounds. If d = cylinder diameter in feet, l = stroke of piston in feet, n = revolutions per minute, and m = number of cylinder ends in operation, the volume capacity of the theoretical compressor will be

$$= \frac{\pi d^2}{4} lnm \text{ cu. ft. per min.}$$

The theoretical horse power, then, is

$$\text{H.P.} = \frac{.7854\, d^2\, lnmW}{33{,}000}. \tag{33}$$

The following examples are intended to bring out the differences in the theoretical work required in isothermal compression as compared with adiabatic, and in the compression of dry air as compared with moist.

Example. — Cyl. dia. $d = 5$ feet; stroke $l = 4.5$ feet; $n = 35$ r.p.m.; compressor is two-cylinder double-acting, so that $m = 4$; suction pressure $= 14$ lbs. per square inch absolute; discharge pressure $= 40$ lbs. per square inch absolute.

Case I. Isothermal Compression, Dry Air. —

From Eq. (10), $W = P_1 \log_e \frac{P_2}{P_1} = 14 \times 144 \log_e \frac{40}{14} = 2130$ ft.-lbs.

$$\text{H.P.} = \frac{.7854 \times 5^2 \times 4.5 \times 35 \times 4 \times 2130}{33{,}000} = 795.$$

Case II. Adiabatic Compression, Dry Air. — Assume $\gamma = 1.408$.

From Eq. (16) $W = P_1 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = 14 \times 144 \times \frac{1.408}{.408}\left[\left(\frac{40}{14}\right)^{\frac{.408}{1.408}} - 1\right]$

$$= 14 \times 144 \times 3.45\,(2.85^{20} - 1) = 2469 \text{ ft.-lbs.}$$

$$\text{H.P.} = \frac{.7854 \times 5^2 \times 4.5 \times 35 \times 4 \times 2469}{33{,}000} = 919.$$

Assuming that the initial temperature of the air was 70° (= 530° abs.), the final temperature will be

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 530\left(\frac{40}{14}\right)^{\frac{0.408}{1.408}} = 717^\circ$$

and the temperature rise $= 717 - 530 = 187^\circ$.

Case III. Compression of Moist Air. — Assume temperature $= 70^\circ$ F.; relative humidity $e = .5$; suction pressure $= 14$ lbs. per square inch absolute; discharge pressure also as before.

From the steam table the partial pressure for the saturated vapor at 70° F. $= .3626$ lbs. per square inch. For $e = .5$, the actual partial pressure then is $P_w = .5 \times .3626 = .1813$ lbs. per square inch. From this $P_a = P - P_w = 14 - .1813 = 13.818$ lbs. per square inch.

Hence $$m = .621 \frac{P_w}{P_a} = .621 \frac{.1813}{13.818} = .0082.$$

Assume for air, $C_p' = .238$, $C_v' = .169$; for water vapor, $C_p'' = .453$, $C_v'' = .345$.

Then

$$\text{mean } \gamma = \frac{.238 + .0082 \times .453}{.169 + .0082 \times .345} = \frac{.2417}{.1718} = 1.406.$$

With this value of γ, the work of compressing one cubic foot of moist air is

$$W = P_1 \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] = 14 \times 144 \times \frac{1.406}{.406}\left[\left(\frac{40}{14}\right)^{\frac{.406}{1.406}} - 1\right]$$

$$= 14 \times 144 \times 3.46\,(2.85^{288} - 1) = 2455 \text{ ft.-lbs.}$$

$$\text{H.P.} = \frac{.7854 \times 5^2 \times 4.5 \times 35 \times 4 \times 2455}{33{,}000} = 914\,.$$

413. Effect of Clearance, Volumetric Efficiency, and Slip. — The fact that there is clearance in the cylinder and that the air caught in this clearance reëxpands, does not in any way affect the work formulas for pounds or for cubic feet of air handled. Its only effect is to reduce the capacity of the compressor.

In Fig. 574, let the volume in the cylinder at the end of the suction stroke = V_1, the volume after compression = V_2, and the clearance volume = V_3. At the end of the instroke the clearance volume V_3 is filled with air under pressure = P_2. On the next outstroke the air expands, following the piston, until at the suction pressure P_1 the volume V_4 is reached. Not until this condition is attained can the cylinder draw in a fresh supply of air from the outside.

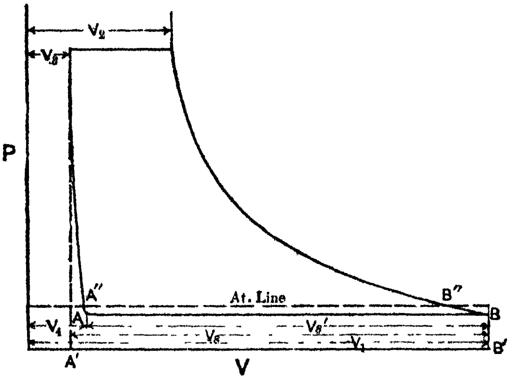

FIG. 574.

Volumetric efficiency, E_v, is defined as the ratio of the volume of free air actually compressed in a given time to the piston displacement in the same time. If multistage compression is used, piston displacement, of course, refers to the low-pressure cylinder only. Engineers use several methods of computing volumetric efficiency, but there is only one accurate way, and that is to determine it experimentally by actually measuring the air taken in, or discharged, or both, as a check. The free air thus determined divided by the piston displacement is the true volumetric efficiency, E_{vt}.

The best approximate computation is the following: Let P_b = the pressure of the free air outside of the cylinder (= barometer pressure); P_1 = pressure in the cylinder at end of suction stroke (less than P_b on account of losses through the intake valves); T_b = absolute temperature of free air outside of cylinder; T_1 the absolute temperature at the end of the suction stroke (usually higher than T_b on account of heating of the incoming air by the cylinder walls); V_s = piston displacement per stroke; $V_{s'}$ = useful or effective piston displacement per stroke (see Fig. 574); G_s = the weight of free air in the volume V_s; and $G_{s'}$ = the weight in the volume $V_{s'}$. Then *approximate*, or *apparent*, volumetric efficiency

$$E_{va} = \frac{G_{s'}}{G_s} = \frac{\dfrac{P_1 V_{s'}}{RT_1}}{\dfrac{P_b V_s}{RT_b}} = \frac{P_1 V_{s'} T_b}{P_b V_s T_1}. \qquad \text{I} \qquad (34a)$$

Of the factors in this equation, P_1 may be measured from an indicator diagram, P_b is found by barometer, V_s is proportional to the distance $V_1 - V_3$ (Fig. 574), and $V_{s'}$ to the distance $V_1 - V_4$. T_b is easily found, but the value of T_1 is uncertain and can only be assumed. On account of this uncertainty, the ratio $\frac{T_b}{T_1}$ is sometimes neglected or assumed equal to 1.0, and the equation for apparent volumetric efficiency is, then,

$$E_{va} = \frac{P_1 V_{s'}}{P_b V_s}. \qquad \text{II} \qquad (34b)$$

This form of the equation corrects for pressure differences, but neglects temperature effects; it is equivalent to the ratio of the distances $\frac{A''B''}{A'B'}$, Fig. 574. Finally, in rough computations a further approximation is made, neglecting $\frac{P_1}{P_b}$, so that

$$E_{va} = \frac{V_s'}{V_s} = \frac{V_1 - V_4}{V_1 - V_3} = \frac{AB}{A'B'}. \qquad \text{III} \qquad (34c)$$

There is, however, no good reason for ever using form III. Form II is largely employed in estimating capacity, *but none of these forms should be used in computing air delivered on tests.* Even assuming that T_1 is known, form I is approximate, in that it neglects losses through leakage. On all tests the air taken in or discharged should be actually measured.* Where computations *must* be based on apparent volumetric efficiencies, the lower part of the diagram should be taken with a weak spring.

Slip is defined as the difference between the piston displacement and the actual volume of free air delivered, both based on the same time unit. The *percentage of slip* is the cubic feet of slip in a given time divided by the piston displacement in the same time.

414. The Efficiency of Compression and the Mechanical Efficiency of the Machine. — The efficiency of compression is the ratio of the theoretical work W_{th} required to compress *isothermally* a given volume of free air under stated conditions from a pressure P_1 to a pressure P_2, to the work W_a required to do this in an actual case. The efficiency of compression, therefore, is

$$E_c = \frac{W_{th}}{W_a}. \qquad (35)$$

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* It is perfectly easy to obtain high apparent volumetric efficiencies, that is, as computed from the cards. With mechanically operated intake valves, for instance, it is possible, by opening the intake at the inner end of the stroke, allowing the air compressed in the clearance spaces to flow into the intake pipe, to obtain a vertical drop from receiver to intake pressure, as far as the indicator diagram is concerned. It must be apparent that this scheme has not in any way improved the true volumetric efficiency of the compressor. Another method, used in an attempt to decrease the effect of clearance, is to equalize the pressure on both sides of the piston near the end of the stroke by having the piston open a by-pass connecting the two ends of the cylinder. This method, while it helps the end of the cylinders in which the pressure is being relieved, must decrease the effective charge volume on the other side, and the gain as far as volumetric efficiency is concerned is, therefore, fictitious.

W_{th} may be directly computed from Eq. (9) or (9a), while W_a is found from actual indicator diagram.

The *mechanical efficiency* E_m of an air compressor is defined as the ratio of the air horse power to the horse-power input used to operate the compressor. If the prime mover is a steam engine, as is most often the case, the power input is the I.H.P. of the steam cylinder or cylinders, if the engine is direct-connected. If any prime mover used to run the compressor is belt-connected, some allowance for the efficiency of transmission must of course be made. This point should be made a matter of specification in every guarantee contract, as the proper allowance to be made for the efficiency of a belt drive is not at all definite, but depends upon circumstances. Ninety-five per cent at full load is a common figure.

The *air* or *compressor horse power* is computed from indicator diagrams, speed, and engine constants in exactly the same way as for the indicated horse power of a steam engine.

It should be noted that the mechanical efficiency of the machine as above computed not only takes into account the friction losses in the machine members, but also the fluid friction losses incident to taking in and discharging the air.

If the ideal isothermal compression line is drawn in on the average compressor diagram of a test, as is done in Fig. 575, some of these losses become apparent and may be estimated.* Thus, in Fig. 575,

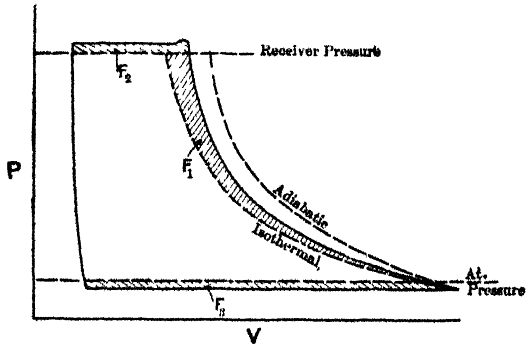

FIG. 575.

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* For a detailed mathematical discussion of the estimation of the various losses in an air compressor, the student is referred to v. Ihering, " Die Gebläse."

Shaded area F_1 = Loss due to heating of air during compression;
Shaded area F_2 = Loss due to work required to force the air (and in some cases also cooling water) through discharge valves and ports;
Shaded area F_3 = Loss due to resistance encountered by the incoming air in passing intake valves and ports.

415. Multistage Compression. — It has already been pointed out that one of the main reasons why the actual work of compression is greater than the theoretical isothermal work lies in the fact that the air is imperfectly cooled during compression. This is one of the practical difficulties that cannot be overcome by any system of cylinder cooling. To minimize this loss as far as possible, especially for high-pressure work, it is common practice to divide the total work into stages and to cool the air between the stages; thus a two-stage compressor consists of a low-pressure cylinder, in which the air is compressed from atmospheric pressure P_1 to some pressure P_2 (see Fig. 576). It is then transferred to an intermediate receiver, called an " intercooler," in which it is cooled more or less, depending

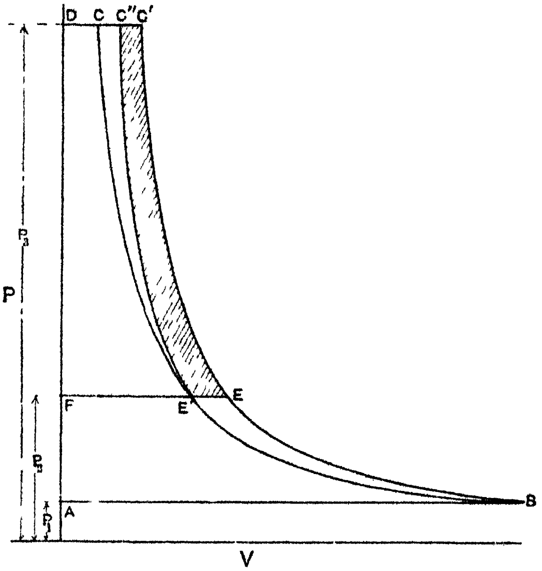

Fig. 576.

upon the efficiency of the cooler. The ideal case, of course, is to cool the air back to the temperature T_1, which it possessed at the beginning. The air from the intercooler next passes into the second or high-pressure cylinder, in which the pressure is raised from the intercooler pressure P_2 to the discharge pressure P_3. Similarly, a three-stage compressor has a low, an intermediate, and a high-pressure cylinder with intercoolers between the cylinders.

How many stages can be economically employed depends largely upon the pressure to which the air is to be raised. It is a fact that each stage will show a saving in work as far as the compression itself is concerned, but there is a point at which such saving is balanced by increased friction and leakage losses and beyond which it does not pay to increase the number of stages. Two-stage compression is commonly employed where the pressure to be attained exceeds 60 to 80 lbs. per square inch; the use of three stages is not very common. Even in two-stage compression, bad valve design and inefficient intercooling may easily make the two-stage operation *less* economical than single-stage between the same pressure limits would have been.

Confining the discussion to two-stage compression, and considering intercooling to initial temperature T_1 the ideal case, the theoretical saving over single-stage adiabatic compression can be easily shown. In Fig. 576, BC is the isothermal and BC' the adiabatic compression line. In single-stage compression the loss due to the absence of cooling is measured by the area CBC'. If now we compress in a low-pressure cylinder to some intercooler pressure P_2 along the adiabatic BE, the work done in the low-pressure cylinder will be represented by the area $ABEF$. The intercooler is assumed to cool at constant pressure P_2 to the initial temperature T_1, locating the point E'. The volume represented by FE' is then drawn into the high-pressure cylinder and compressed along the adiabatic EC'' to the pressure P_3, the work done in this cylinder being $FE'C''D$. The total work done in both cylinders is represented by the area $ABEE'C''D$, which is less than the work $ABC'D$ by the area $EE'C''C'$, which therefore represents the saving under the conditions assumed.

The work done in the two cylinders should be equal, which means that there is some theoretically best intercooler pressure P_2 which should be maintained. The student is referred to Peele, " Compressed Air Plant," for the development of mathematical expressions for the work done in each cylinder and for the best intercooler pressure.

416. The Construction of the Combined Diagram for a Two-stage Compressor and the Saving due to Cylinder Jackets and Intercooler. — In practice the discussion of the previous article is modified by the following factors: The compression in each cylinder is not adiabatic, but, owing to the use of cylinder cooling, the compression line lies between the adiabatic and the isothermal; the action of the intercooler may not be effective enough to cool the air to the initial temperature, or if the cooling is very efficient the temperature may be less than T_1 at entrance to the high pressure; owing to losses in passing through valves and intercooler, the diagrams for the two cylinders may overlap instead of meeting in a line like FE', Fig. 576. The overlapping area means lost work. Finally, the clearances in the two cylinders, which are usually not the same percentage of the stroke volumes, has a certain effect upon the relative position of the cards.

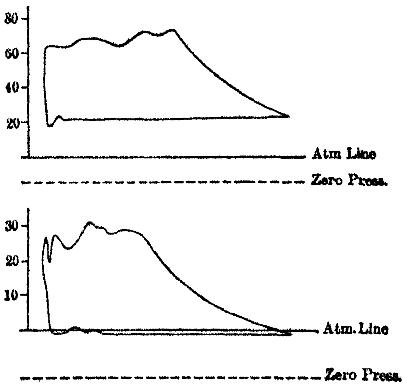

FIG. 577.

To construct the combined diagram, select average high- and low-pressure cards from the set obtained on the test, and proceed to combine these in exactly the same way as explained for steam-engine diagrams in Chap. XVI, setting off the diagram for each cylinder from the zero volume line a distance proportional to the clearance for that cylinder. Fig. 577 shows the high- and low-pressure cards for a two-stage compressor, for which the following table contains the main data:

Diameter, H.P. cylinder..........	24.25″
Diameter, L.P. cylinder..........	38.5″
Stroke..........................	48″
Piston displacement, H.P. cylinder..	12.83 cu. ft.
Piston displacement, L.P. cylinder..	32.34 cu. ft.
R.p.m.........................	72
Clearance, H.P.cylinder..........	1 per cent.
Clearance, L.P. cylinder..........	1 per cent.
Clearance volume, H.P. cylinder...	.128 cu. ft.
Clearance volume, L.P. cylinder....	323 cu. ft.
Total volume, H.P. cylinder.......	12.958 cu. ft.
Total volume, L.P. cylinder........	32.663 cu. ft.
Barometer reading, 28.6″ =.......	14.0 lbs. absolute.
Intercooler pressure, 26 lbs. =.....	40.0 lbs. absolute.
Reservoir pressure, 65 lbs. =.......	79.0 lbs. absolute.

Fig. 578 shows the two diagrams combined. To show the saving effected by the low-pressure jacket, draw from *E* the isothermal *EC* and the adiabatic *EG*. Since, without any cooling whatever, the compression line would have followed *EG*, the area *EGFE* will represent the saving due to the low-pressure jacket. Assume that, beginning with *F*, the air is discharged into the intercooler at constant pressure. The volume discharged is represented by *FX*. Now, *if the reëxpansion lines of the two diagrams crossed the intercooler pressure line at the same point, and if there were no intercooling whatever*, the suction line of the high-pressure card would bring us back exactly to the point *F*. But there is usually a difference in the volumes shown by the reëxpansion at intercooler pressure, owing to differences in clearance in the two cylinders, which difference in this case is equivalent to the distance *XD*. Hence, still assuming that there is no intercooling, the high-pressure suction line will extend only to *F′* instead of to *F*. Adiabatic compression in the high-pressure cylinder would then have given the line *F′I*. Intercooling, however, has decreased the volume at the end of high-pressure suction to the point *K*, and adiabatic compression from that point gives the line *KL*. Hence the saving due to intercooler action is measured by the area *F′ILK′F′*. From *K* draw also the isothermal *KN*. The area *KLMK* included between the adiabatic *KL* and the actual compression line, *KM*, measures the saving due to the high-pressure jacket.

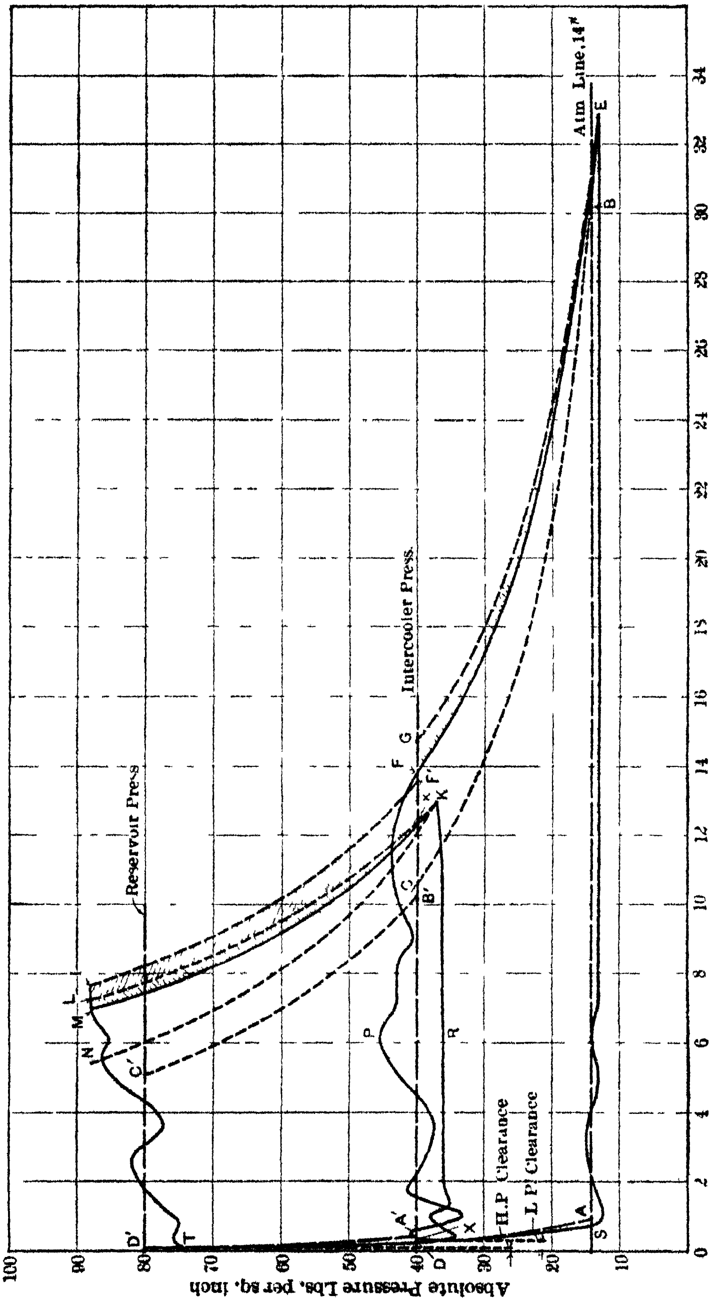

FIG. 578.—COMBINED DIAGRAM FOR AIR CYLINDERS, AIR COMPRESSOR TEST.

The combined diagram here given is not a good example of practice in two-stage compression. This will become apparent when we compare the actual gains made with the possible gains and actual losses. The possible saving in cooling the cylinders is measured by the area *EGCE* in the low-pressure, and by area *KLNK* in the high-pressure cylinder. Both cylinder jackets, therefore, appear to be inefficient. The intercooler, when efficient, should cool the air back to the initial temperature, that is, back to the isothermal *EC*. Its inefficiency is consequently apparent. The actual friction and valve losses are in this diagram indicated by the area below the atmospheric-pressure line, the area above the reservoir-pressure line, and the overlapping area of the two cards, above and below the intercooler-pressure line. It may be assumed that two of these losses, that at low-pressure suction and high-pressure discharge, are present also to a similar extent in single-stage compression; but the loss, due to the friction in low-pressure discharge valves, intercooler, and high-pressure suction valves, measured by the overlapping area *KPXRK*, is directly chargeable to the use of the two-stage principle and should be offset against the intercooler saving. In this case, the net result is a positive loss, owing to inefficient intercooling and bad design of intercooler and valves.

417. The Computation of Efficiencies in Multistage Compression. — The true volumetric efficiency E_{vt} and the mechanical efficiency E_m are computed exactly as for the single-stage compressor (see Arts. 413 and 414). The apparent volumetric efficiency E_{va} is, of course, computed from the low-pressure card. The efficiency of compression E_c is again the ratio of the theoretical (isothermal) work of compression to the actual work. In Fig. 578, the actual work is the equivalent of the sum of the areas of the two cards, that is, *EFPXSE* and *KMTDRK*. The theoretical work may either be computed from the equation given in Art. 411, or it may be determined from the combined diagram. The area representing the theoretical work for the low-pressure cylinder is *ABCXA*, the line *XA* being the isothermal reëxpansion line. To determine the theoretical work for the high-pressure cylinder, set off the distance *CB′* equal to the distance *DX*, and draw the isothermals *B′C′*

and $D'A'$. The theoretical work for the high-pressure cylinder is then represented by the area $A'B'C'D'A'$.

The higher the efficiency of cooling, the greater, of course, will be the value of E_c. With a very efficient intercooler and low temperatures of cooling water, it is even possible to cool to such an extent that the point K will lie to the left of the isothermal EC, in which case the efficiency of compression may approach or exceed 100 per cent, provided, of course, that friction losses are reduced to the minimum.

418. Arrangements for Testing Piston Compressors. — The main items to be determined for an efficiency and economy test are: (A) the power input (steam horse power, motor horse power, etc.); (B) the air horse power; and (C) the weight or volume of air handled. A complete test should further record speed, pressure, and temperature of surrounding air, inlet and outlet pressures and temperatures, and all intermediate pressures and temperatures, if multi-stage compression is used; also the temperatures of all the water entering and leaving all of the jackets.

A. The Power Input. — The most common motive power is steam; next electric energy is largely employed. For methods of determining the power delivered to the air cylinders by any of the types of prime movers ordinarily employed, see the particular chapters concerned.

When the prime mover is an electric motor, hydraulic motor, or when a belt drive is used, the measurements on this end confine themselves generally to the determination of the horse-power input. In case a steam engine is used, the test may of course be extended to a complete test of engine and boiler plant, for which see Chaps. XVII and XVIII.

B. Air Horse Power. — This is the indicated horse power as computed from cards taken from the air cylinders. The computation is exactly the same as for horse power from steam cylinders.

The rules to be observed in the attachment and use of indicators on the air cylinders are also the same as for steam (see Chaps. XV and XVI), and no special precautions are necessary.

C. Weight or Volume of Air Handled. — This measurement is very important and is the one hardest to make accurately in connection with an air-compressor test, especially if the plant is large. In that case some of the most reliable means available, that is, gasometer and meter, can no longer be used on account of the cost of the apparatus. In fact, under any circumstances, the measurement is not a simple one, and for that reason it is often agreed to compute the quantity of air delivered from the piston displacement and the apparent volumetric efficiency (as computed from the card). It should be distinctly understood that this method is approximate and may lead to considerable errors. If used, E_{va} should be computed only from a card taken with a weak spring and stops.

For very accurate work, air may be measured both at inlet and outlet of the compressor. The means available at the inlet are: gasometer, meter, anemometer. Of these the former is the most accurate. The anemometer is not recommended, although it possesses the decided advantage of large capacity as compared with the other two. At the outlet, besides the three methods above given for the inlet, any one of the following methods is also available: orifice, Venturi meter, Pitot tube, nozzle, pumping up a tank from a lower to a higher pressure, calorimetric means.

The theory underlying any of these methods, together with the accuracy attainable and the limitations to applicability in individual cases, has been discussed at length in Chap. XII. One or two illustrations of actual apparatus used may, however, be of service.

Fig. 579 * shows the method employed by Professor Josse in the laboratory at Charlottenburg for small-sized air compressors. The compressor C is in this case driven by belt from a steam engine D. The air is measured both at inlet and outlet. G is a gas meter connected to the inlet; R_1 being a small receiver to deaden the pressure fluctuations due to suction, and V_1 a valve to control the suction pressure if desired. Manometers are indicated to observe the pressures both at the meter and in the suction receiver. The high-pressure air is first delivered into a small receiver R_2 and from this is either allowed to escape at O or is pumped into the large measuring tank R_3. The valve V_2 serves to regulate the pressure,

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* *Zeitschrift des Vereins deutscher Ingenieure,* Feb. 4, 1905.

shown by gauge G_2, against which the compressor is discharging, and this is kept constant during a trial of air delivery. The capacity of the large tank is about 700 cubic feet. The method of operating this apparatus is practically that outlined in Art. 230, p. 438.

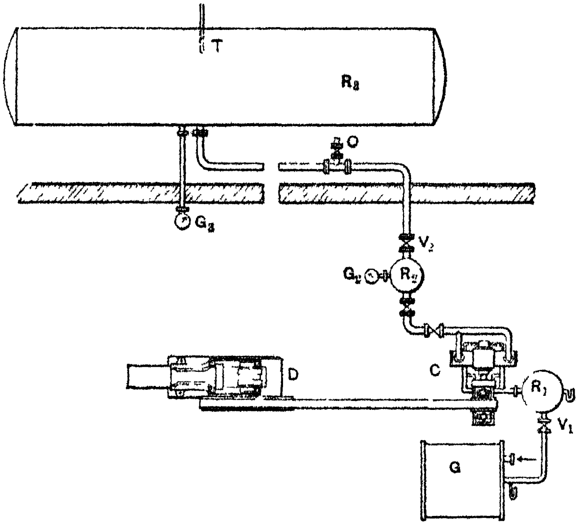

Fig. 579.—Method of Measuring Air by Pumping up a Reservoir.

Air is allowed to escape at O, until the pressure at G_2 and that in the tank, measured at G_3, are determined. Observe also the temperature at T. Then, closing O, let the pressure at G_3 increase to the desired limit, keeping G_2 constant. At the end observe G_3 and the temperature. The formula for computing weight of air from this data is also given in the article mentioned.

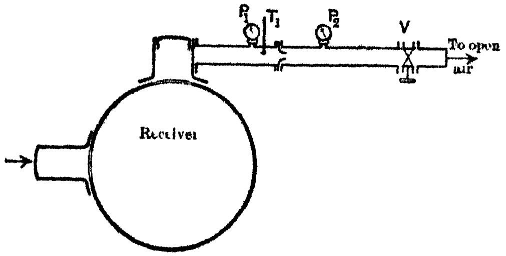

Fig. 580.—Orifice Method of Measuring Air at Outlet from Compressor.

Fig. 580* shows an orifice method of measurement used in the test of a compressor delivering about 140,000 cubic feet of air per hour against a pressure of about 115 lbs. The compressor delivered first into a receiver having a capacity of 1050 cubic feet. The interposition of a receiver is practically a necessity for any method of measurement, to obtain steady flow. The orifice used was bell-mouth, diameter at small end 1.42″, large end 2.32″, length 2.16″, radius of curvature at entrance 80″, constant $C = .975$. For discharge formula see Arts. 220 to 223. The arrangement of apparatus for measuring pressure and temperature are indicated. V is a regulating valve to maintain at the proper level the pressure P_1 against which the compressor is discharging.

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* *Zeitschrift des Vereins deutscher Ingenieure*, Oct. 30, 1909.

Another orifice method, capable of large capacity, is shown in Fig. 581. Here the compressor delivers into the receiver R_1, in which the desired pressure is maintained by means of valves V_1 and V_2, as read at G_1. The air discharges into R_2, which is furnished with a number of discharge orifices around the bottom as indicated. The pressure in R_2 is determined by the open mercury column M, which may also be used to measure the pressure in R_1 by throwing over the three-way cock H, unless the pressure is too high. Temperature is measured at T. The water seal merely serves the purpose of deadening noise. By substituting plugs for the orifices the capacity of the apparatus may be changed at will. The computation of delivery is of course the same as for any orifice measurement.

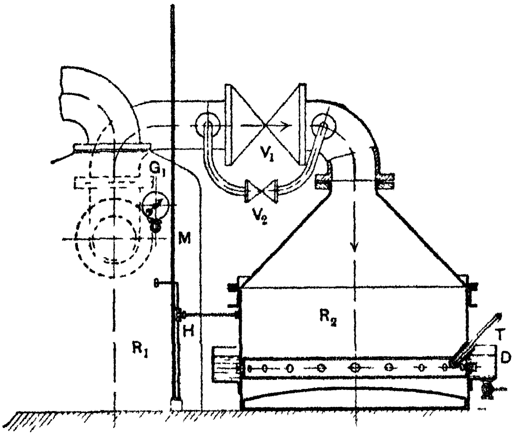

Fig. 581.—Orifice Method of Measuring Air, for Large Capacities

419. Record of Data and Computations for Air-compressor Test. — The forms on pages 979 and 980 show the necessary data to be observed for a test on a two-stage compressor driven by a compound engine. The method of air measurement is in this case by convergent nozzle. The pressure discharged against by the compressor is kept at the desired point by a valve just ahead of the nozzle. The first form takes care of the data and computations obtained from the steam and air cards. The second records the general readings of speed, pressures, temperatures, etc.

The principal data and results to be reported are given in the following table:

RESULTS OF AIR-COMPRESSOR TEST.

1. Date of test ... .
2. Duration of test, hours.......... ... .
3. Diameter of cylinders, inches:
    (*a*) Steam, H.P... ., (*b*) Steam, L.P.........;
    (*c*) Air, H.P. .....; (*d*) Air, L.P.........
4. Stroke, inches:
    (*a*) Steam cylinders ... ...; (*b*) Air cylinders........
5. Diameter piston rods, inches ....
    Av. per cent of clearance in air cylinders, H.P.........; L.P.. .
    Revolutions per minute:
        (*a*) Prime mover . .....; (*b*) Air cylinders . .. .
6. Average observed readings:
    (*A*) Pressure, absolute:
        (*a*) Steam .. . ; (*b*) Vacuum, condenser....; (*c*) Barometer . ; (*d*) Air suction ....; (*e*) Intercooler .. . ;
        (*f*) Receiver......; (*g*) Pressures at measuring apparatus. . .
    (*B*) Temperatures, °F.:
        (*a*) Steam ... .; (*b*) Air at intake......; (*c*) Air leaving L.P. cyl.......; (*d*) Air leaving intercooler. ...; (*e*) Air leaving H.P. cyl......; (*f*) Jacket water, L.P. cyl., entering .. . ; leaving......; (*g*) Jacket water, H. P. cyl., entering. . . ; leaving.. ...; (*h*) Jacket water, intercooler, entering. ... ; leaving......; (*i*) Temperatures at measuring apparatus.....
    (*C*) Weights, pounds total:
        (*a*) Condensed steam, or feed water to boiler........; (*b*) Steam to steam jackets. ....; (*c*) Cooling water, L.P. air-cyl. jacket ......; (*d*) Cooling water, intercooler jacket......; (*e*) Cooling water, H.P. air-cyl. jacket........

RESULTS OF AIR-COMPRESSOR TEST.—*Continued.*

7. I.H.P., H.P. steam cylinder ....
8. I.H.P., L.P. steam cylinder ....
9. Total I.H.P., steam end ....
10. I.H.P., H.P. air cylinder ....
11. I.H.P., L.P. air cylinder ....
12. Total air I.H.P. ....
13. Pounds of wet steam per steam I.H.P. per hour ....
14. Quality of steam, per cent ....
15. Pounds of dry steam per steam I.H.P. per hour ....
16. Pounds of dry steam per air I.H.P. per hour ....
17. Heat supplied in one pound of steam, B.t.u. ....
18. Heat supplied per hour per steam I.H.P., B.t.u. ....
19. Heat supplied per hour per air I.H.P., B.t.u. ....
20. * Actual volume of *free* air delivered per hour, cu. ft. ....
21. Piston displacement of L.P. cylinder, per hour, cu. ft. ....
22. Theoretical horse power required to deliver the actual volume of free air compressed to receiver pressure, adiabatic compression ....
23. Theoretical horse power required to deliver the actual volume of free air compressed to receiver pressure, isothermal compression ......
24. Slip, cubic feet per hour ....
25. Slip, per cent. ....
26. Efficiencies:
    - (*a*) Mechanical efficiency, E_m ....
    - (*b*) Volumetric efficiency, true or actual, E_{vt} ....
    - (*c*) Volumetric efficiency, apparent, E_{va} ....
    - (*d*) Efficiency of compression, E_c ....
27. Saving effected by cooling jackets, as compared with adiabatic compression and no intercooling; computed from combined diagram:
    - (*a*) Saving due to L.P. jacket, horse power ....
    - (*b*) Saving due to H.P. jacket, horse power ....
    - (*c*) Saving due to intercooler jacket, horse power ....

In consideration of the explanations given in previous paragraphs of this chapter, none of the items of the above table need any extended comment. For items 22 and 23, see Arts. 411 and 412. Items 24 and 25 are explained in Art. 413. For the computation of the efficiencies, see Arts. 413 and 414.

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* Free air means air under the conditions of barometer pressure and air temperature existing at the intake to the compressor. The item is computed from the data observed on the measuring apparatus used, and full details of this apparatus and the observations made should be given in the report. It has already been stated that if no actual measurement is made, the quantity may be taken equal to piston displacement multiplied by apparent volumetric efficiency.

SIBLEY COLLEGE, CORNELL UNIVERSITY.

Air Compressor built by... at ...
Tested at... Date.. 191...
Cards integrated by.. Checked by...
Scale of springs ... Steam high press . Steam low press.. ..Air high press .. Air low press ...

DIMENSIONS.

Steam Cylinders.

High Pressure.		*Low Pressure*	
Diam. in inches		Diam in inches	
Area in sq in....		Area in sq in	
Diam piston rod in in		Diam. piston rod in in.	
Area in sq. inches		Area in sq. inches	
Length of stroke in feet.		Length of stroke in feet	

Piston Displacement in Cubic Feet.

Head ..	Crank	Head ...	Crank.

Volume in Clearance Per Cent.

Head..	Crank .	Head .	.Crank .

Air Cylinders.

High Pressure.		*Low Pressure.*	
Diam in inches...		Diam. in inches ...	
Area in sq in		Area in sq in. ...	

Diameter of Piston Rods in Inches.

Head ...	. Crank ..	Head..	..Crank...

Area of Piston Rods in Square Inches.

Head .	..Crank. ...	Head ..	Crank...
Length of stroke in feet. ..		Length of stroke in feet..	

Piston Displacement in Cubic Feet.

Head .	Crank. ...	Head ...	Crank. ...

Volume in Clearance Per Cent.

Head	Crank . .	Head . .	Crank...

	Steam Engine Diagrams									Air Cylinder Diagrams.								
No.	Rev per Min.	Area	Length.	M E P	I H P	Area	Length.	M.E P.	I H.P	Rev per Min.	Area	Length.	M.E.P.	I.H P.	Area.	Length.	M.E P	I.H.P.
1																		
2																		
3																		
4																		
5																		

MECHANICAL LABORATORY, SIBLEY COLLEGE, CORNELL UNIVERSITY.

Test of.... ....Air Compressor at.. . ...Date... 191. ....

Diam. of Air Nozzle in inches.. . . Area of Air Nozzle in sq. ft . .. Coefficient Air Nozzle.. ....

		Revolutions		Gauge Readings			Temperatures													Heat Absorbed by Jacket Water.			
					Air		Air							Jacket Water.									
Number.	Time	Continuous Counter	Per Minute.	Boiler or Steam-chest Pressure.	Reservoir Pressure.	Nozzle Pressure.	Entering Low-pressure Cyl.	Leaving Low-press. Cyl.	Entering High-press Cyl	Leaving High-press Cyl.	Nozzle.	Outside.	Calorimeter	Entering Cooler.	Leaving Cooler or Entering Low-press. Cyl.	Leaving Low-press. Cyl. or Entering High-press. Cyl.	Leaving High-press Cyl	Weight of Jacket Water.	Weight of Condensed Steam.	From Cooler.	From Low-press. Cyl.	From High-press. Cyl.	Total.
				P^1	P^2	P^3	T^1	T^2	T^3	T^4	T^5	T^6	T^7	T_a	T_b	T_c	T_d						
1																							
2																							
3																							
4																							
5																							
Etc.																							
Average.																							

420. The Action of the Centrifugal Fan. — A centrifugal fan, of whatever type, takes gas having a pressure of p_1 pounds absolute, a velocity of v_1 feet per second, and a temperature of $t_1°$, and compresses it to a pressure of p_2 pounds absolute, with a velocity of v_2 feet per second, and a temperature of $t_2°$. In practice the degree of compression is generally so small that there is no sensible development of heat, and consequently the compression may be considered isothermal. The work done by the fan upon the gas is expended in three ways: (*a*) compressing from the pressure p_1 to the pressure p_2; (*b*) forcing the weight of air handled against the pressure p_2; and (*c*) raising the kinetic energy of the gas by increasing the velocity.

The work of compressing the gas is, under ordinary conditions, very small. Thus, if the pressure is raised to 12″ water above atmosphere, the percentage of pressure increase, assuming atmospheric pressure at 14.7 lbs., is only $\frac{.43}{14.7} = .029 = 2.9$ per cent. It is therefore usual to neglect this work item, that is, to assume that the density of the gas is not changed.

This modification, or assumption, however, makes the condition of operation for the fan the same as for a centrifugal pump, except that air or gas instead of water is the fluid pumped. The centrifugal pump theory is discussed in Chap. XXV, and it is there shown that this can be based upon that for the radial outward-flow reaction turbine. The fundamental theory of this turbine therefore applies directly to the centrifugal fan.

It is shown (Art. 471) that the theoretical lift or head produced by the centrifugal pump may be expressed by

$$H = \frac{u_2(u_2 + w_2 \cos \alpha_2)}{g} \text{ feet,} \tag{36}$$

in which u_2 = the linear velocity, in feet per second, of the fan blade at the tip (exit);

w_2 = the relative velocity, in feet per second, of the air leaving the fan wheel (= the volume Q of air flowing, in cubic feet per second, divided by the peripheral area of the fan wheel in square feet, around the exit circumference); and

α_2 = the angle made by u_2 (drawn perpendicular to the radius from the center of the wheel) and w_2 (drawn tangent to the last blade element). See Fig. 611, p. 1054.

It is also shown (p. 1075) that the general equation for gross head h pumped through by a centrifugal pump (or produced by a centrifugal fan) is

$$h = \left\{ h_d - (\pm h_s) + h_1 + \frac{v_d^2 - v_s^2}{2g} \right\} \text{feet}, \qquad (37)$$

in which h_d = static head above atmosphere measured at some point in the discharge pipe;

h_s = static head above atmosphere measured at some point in the suction pipe (h_s is to be used with the minus sign when the pump lifts water, with the plus sign when the water is supplied under some pressure);

h_1 = vertical height between the points of measurement in discharge and suction pipe;

$\frac{v_d^2}{2g}$ = velocity head in the discharge pipe at point where h_d is measured; and

$\frac{v_s^2}{2g}$ = velocity head in the suction pipe at point where h_s is measured.

It is needless to say that for the fan all these heads are expressed in feet of air or gas.

The above equation applies, as it stands, to the case of a fan drawing from a closed space and discharging into another limited space. That is not the usual case. Usually the fan either draws from the air and discharges into a limited space, or it draws from a limited space and discharges into air.

In the case of the fan drawing from the atmosphere, both h_s and v_s may be considered zero. There is, of course, velocity near the inlet of the fan, but where velocity exists, h_s is no longer zero.

Away from the fan, however, h_s and v_s are zero. The gross head then becomes

$$h = \left(h_d + h_1 + \frac{v_d^2}{2g}\right) \text{feet.} \tag{38}$$

In practice h_1 is neglected, so that finally

$$h = \left(h_d + \frac{v_d^2}{2g}\right) \text{feet.} \tag{39}$$

Where the fan discharges into air, h_d and $\frac{v_d^2}{2g}$ may be considered equal to zero, and, neglecting h_1, the gross head in that case becomes equal to

$$h = \left(h_s - \frac{v_s^2}{2g}\right) \text{feet.} \tag{40}$$

421. Work done by a Fan: Air Horse Power. — It has already been stated that work done by a fan consists of

(*a*) the work of compressing the gas from suction pressure to discharge pressure. It has been shown that this is a small quantity and is usually neglected.

(*b*) the work of forcing the gas against the discharge pressure. This work is equivalent to lifting the weight of gas handled through a height corresponding to the difference of static head on the two sides of the fan.

(*c*) the work of increasing the velocity of the gas flow from the velocity on the suction to the velocity on the discharge side.

Take the case of a fan drawing from the atmosphere and discharging through a main against a resistance. At any section *A* of the discharge main, Fig. 582, let the static pressure be equal to p_d (measured above atmosphere, in inches of water, mercury, ounces per square inch, or other units), and let the velocity be v_d feet per second. Also let Q be the volume in cubic feet of gas delivered per second, and δ be the density of the gas (= weight per cubic foot at the pressure and temperature in the discharge main).

Compute the equivalent height of gas column h_d in feet corresponding to the pressure p_d (see Art. 233). Then, since the weight

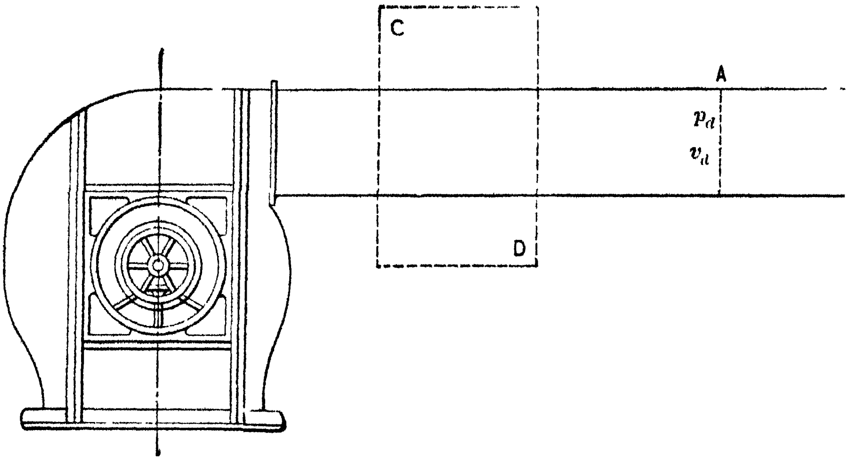

FIG. 582.

of gas handled is $G = Q\delta$ pounds per second, and since this weight is raised against the pressure p_d (= equivalent to the height h_d), the work done under heading (*b*) above will be

$$W_b = Gh_d = Q\delta h_d \text{ ft.-lbs. per sec.} \qquad (41)$$

The general formula for kinetic energy is

$$KE = M\frac{v_2^2 - v_1^2}{2} = \frac{G}{g}\,\frac{v_2^2 - v_1^2}{2} = G\frac{v_2^2 - v_1^2}{2g} \text{ ft.-lbs.,} \qquad (42)$$

in which v_1 *is the velocity of the mass* M *at the beginning, and* v_2 the velocity at the end. In this case, since the fan is drawing from the atmosphere, $v_1 = v_s = 0$, and $v_2 = v_d$. Hence the work equivalent to the kinetic energy under heading (*c*) above is

$$W_c = G\frac{v_d^2}{2g} = Q\delta\frac{v_d^2}{2g} \text{ ft.-lbs. per sec.} \qquad (43)$$

The total work done by the fan upon the gas, therefore, is

$$W = W_b + W_c = Q\delta\left(h_d + \frac{v_d^2}{2g}\right) \text{ ft.-lbs. per sec.} \qquad (44)$$

The last factor of the right-hand member of this equation is equal to the total or gross head by Eq. (39). The foot-pounds of work done in a given time, therefore, is equivalent to the weight in pounds of gas handled in that time, multiplied by the total head in feet.

The result would have been the same if other conditions of operation (discharge into atmosphere or drawing from a closed space and discharging against a resistance) had been assumed, and the general work formula is (neglecting h_1, as is usual)

$$W = Q\delta\left(h_d + h_s + \frac{v_d^2 - v_s^2}{2g}\right) \text{ ft.-lbs. per sec.} \tag{45}$$

Returning to the fan of Fig. 582, a simpler way of obtaining the necessary data for the computation of work done would have been to determine at once the total head at A by using an impact tube (Pitot tube without the static opening). The reading of the instrument is then directly equivalent to the sum of static and dynamic heads $\left(= h_d + \frac{v_d^2}{2g}\right)$. In making this measurement, traverses of the main must be made to obtain the average reading.

Another method of determining the total head against which the fan works is to construct a large box or receiver just beyond the outlet of the fan (CD in Fig. 582), so that the velocity v_d across this box is negligible. The total head produced by the fan then exists as static head, and this may be found by simply connecting a manometer to the box. It is, however, not particularly easy to get a box large enough and to so baffle it that the air currents in it do not manifest themselves sufficiently to affect the manometer reading, and for that reason the determination of total head by impact is probably just as reliable.

Since one horse power is equivalent to 550 foot-pounds of work per second, any of the formulas (41), (43), (44), and (45), divided by 550, will give the corresponding horse power. Eq. (44) for the fan of Fig. 582, and Eq. (45) for the general case, will give the total horse power of work done by the fan. This is called the *air horse power*.

422. Fan Efficiencies. — (*a*) *Manometric Efficiency* is the ratio of the gross head produced by a fan to the theoretical head, or

$$E_{\text{man.}} = \frac{h}{H}, \tag{46}$$

in which H and h are found by means of the equations given in Art. 420, taking care to use for h the expression applying to the case.

(*b*) *Mechanical Efficiency*, or simply *Efficiency of the Fan*, is the ratio of the total work done by the fan in moving air or gas divided by the horse power input to the fan (not to the prime mover). This efficiency, therefore, equals

$$E_m = \frac{\text{Air horse power}}{\text{Applied horse power}} = \frac{Wh}{AHP \times 33{,}000}, \tag{47}$$

in which W = air or gas handled per minute, and h is the gross head produced in feet.

(*c*) *Volumetric Efficiency** is defined as the ratio between the actual volume of air or gas passing the fan in a given time divided by the volume of the fan wheel multiplied by the number of turns the wheel has made in the same time; that is,

$$E_v = \frac{Q}{V_w n}, \tag{48}$$

in which Q = volume of air or gas handled,
V_w = volume of wheel, and
n = revolutions per minute.

This ratio seems to assume, as a standard, that the fan should discharge a volume of air every turn equal to the volume of the wheel. Not much weight is attached to this efficiency in fan testing.

423. **Arrangements for Testing Fans.** — For a capacity and economy test, the determinations must include the following:

(*A*) Power input; (*B*) power output (air horse power); (*C*) volume of gas delivered (capacity); (*D*) readings of velocity and pressure at inlet and outlet; (*E*) speed of fan; (*F*) humidity and temperature.

(*A*) *Power Input.* — Any type of prime mover may be used; those usually employed are steam engines, electric motors, belt drives from main shafting. The first two may be direct- or belt-connected. Where belt-driven, a direct-connected transmission dynamometer should be employed for very accurate work. When direct-connected to a steam engine or motor, the I.H.P. of the engine or the watts input to the motor corrected for the mechanical efficiency of the machine will be the power input to the fan. If the mechanical efficiency of the prime mover is not known, a curve between mechanical efficiency and power input to the prime mover (I.H.P. or watts) must first be found by disconnecting the fan and taking off power by means of a Prony brake. In the form below this horse-power input is called *applied horse power* (A.H.P.).

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* This term is practically obsolete. "Relative delivery" is a better term for this ratio and is coming into use.

(*B*) *Power Output.* — This is the air horse power and is computed as shown in Art. 421.

(*C*) *Volume of Gas Delivered.* — This is, of course, equal to the product of cross section of tube or main through which the gas is moving by the velocity of passage. For the method of measuring the latter, see next paragraph. Calorimetric methods (see Art. 235) may also be used.

(*D*) *Measurement of Velocity and Pressure.* — The instruments commonly used for measuring velocity are the anemometer, Pitot tube, and the Venturi meter. Orifice methods are also used.

For the theory, characteristics, and method of use of any of these instruments, see Chap. XII.

The anemometer determines the velocity directly. To obtain an average reading, divide the inlet and outlet areas into partial areas by the use of string or fine wire. Obtain the average velocity at the center of each partial area. If the latter are equal, the average velocity is the arithmetical mean of all of these readings. The capacity is to be computed from the velocity and cross sections of the outlet. If this volume is smaller than that computed for the inlet, the difference is due to leakage and back flow. The difficulty of calibrating an anemometer with ease and certainty of results makes this instrument less dependable than some of the others.

The Pitot tube is commonly used in the discharge main only. To determine average velocity head, it is necessary to make very careful traverses of the main. For the method of doing this, see Chap. XII. The ratio of the center velocity to the average velocity for the entire pipe section is known as the *pipe factor*, and for extended investigations with the same apparatus it is common to determine the pipe factor by a preliminary investigation and to set the tube once for all at the center of the main.

It has been shown that the Venturi meter is a satisfactory instrument for measuring flow of gas, and it certainly deserves a more extended trial for this service than has up to this writing been given it. See Art. 234.

Orifices may be placed either across the discharge main some distance back from the outlet, or in the outlet itself. For the conditions to be met with reference to pressure difference and the precautions to be observed in order to obtain reliable results, see Chap. XII.

Static pressure is commonly measured by a simple manometer, which may be of the multiplying type (see Chap. VI) if the pressure is very small. It is of the greatest importance to see that the inner end of the pressure tube is not affected by velocity (impact), and to that end it is necessary not to have the end of the tube project beyond the inside surface of the wall of the main. It is hardly necessary to say that for the purpose of determining total head (for computing air horse power) the static pressure must be measured at the same cross section of the main as the velocity head.

In case the air horse power is determined from the measurement of static pressure in a pressure box (velocity head assumed negligible) it is well to measure this pressure at several points in the box, as it is not at all easy to obtain a uniform transformation of velocity into pressure head, especially near the fan. It will be found that careful baffling is required.

Total pressure head may be determined at any section of the main by traversing with an impact tube, that is, a Pitot tube without the static tube. The manometer reading will then be the sum of the static plus the velocity head (= total head).

(*E*) *Fan Speed.* — This may be determined by any of the forms of counter mentioned in Chap. VIII, the choice of the type of instrument depending of course primarily upon the speed.

(*F*) *Determination of Humidity.* — See Art. 410. The effect of humidity upon the work formulas developed is to change the density δ in the equations. In most cases this is not of sufficient importance to take into account.

424. Scope of Test and the Report. — A complete fan investigation should include the following:

(*a*) Determination of the pressure produced with the fan outlet closed;

(*b*) A series of runs at constant speed, varying the size of outlet opening by predetermined amounts;

(*c*) Repeat the series under (*b*) for a series of speeds maintained constant in each case for the same series of outlet openings.

The observations required have been discussed in the previous paragraph. The following forms for recording the observations and the results of computations apply directly to the fan-testing set used at Sibley College. This consists of a Sirocco fan driven by belt through a direct-current motor. The fan delivers into a receiver or pressure box in which the velocity is theoretically zero. The total head (all pressure head) is here determined by means of a manometer. In the discharge pipe a Pitot tube is used and finally a Venturi meter is connected beyond the Pitot tube. An anemometer is also used. This determines the flow of air, first, by anemometer at inlet and outlet, second, by static pressure head in the pressure box, third, by Pitot tube, and fourth, by Venturi meter. The purpose of the experiment is not only to test the fan but also to give practice in several methods of measuring the flow of gas.

The complete report should show the results graphically as follows: With the series of ratios $\frac{F_0'}{F_0}$ (actual outlet opening to full opening) as abscissas, draw curves with the following ordinates:

(*a*) Efficiency of fan, E_m;

(*b*) Air horse power;

(*c*) Volume of free air delivered in cubic feet per minute. This will give one curve sheet for each speed used.

The first of the following forms shows the scheme used for recording the observations, the second shows the principal items to be computed from the observations.

TEST OF . . . . FAN, AND MEASUREMENT OF FLOW OF AIR.

Log of Test.

Date , Observers . . . . . , ... , ... , ...
Barometer, inches Hg . . . ; Temperature, ° F. . . . . ; Humidity, per cent. . . . .
Calibration Constants: Anemometer. . . . , Pitot Tube . . , Venturi Meter. . . . . , Multiplying Manometer. . . . . , Other Constants. . . . Full Area of Fan Outlet, sq. ft. . . . . (= F_0).

Run Number.	Ratio of Outlet Area F_0' to Full Area F_0 Per cent	Speed, R.P M.		Input to Motor.		Watts lost in Motor, W_L	Manometer Readings.					Anemometer.	
		Motor.	Fan	Volts, E.	Amperes, I.		Pressure Box, h_1.	Pitot Tube.			Venturi, h_5.	Velocity.	
								Static, h_2.	Velocity, h_3.	Impact h_4.		Intake, V_{a1}	Dis-charge, V_{a2}.

TEST OF FAN AND MEASUREMENT OF FLOW OF AIR.

RESULT SHEET.

Tests made by..........., ..., ..., Date........ 191

Notation. h = head in inches of water, h' = head in inches of other fluids; $h = h'K$, where K is the ratio of density of manometer liquid to that of water; $H = hr = h'Kr$ is equivalent head in feet of air, in which r is the number of feet of air column giving same pressure per sq. inch, as 1 inch of water; F = area in sq. ft ; V = velocity in feet per sec ; Q = volume in cubic feet per sec.; W = weight in pounds per sec

Type of fan
Manufacturer
Outside diameter of wheel, ins
Width of wheel, ins
Dimensions of blade, axial, ins
Dimensions of blade, radial, ins
Form of blade
Diameter fan inlet, ins
Area fan inlet, sq ft., F
Diameter fan outlet, ins
Area fan outlet, sq ft., F_0
Diameter discharge pipe, ins
Length of discharge pipe, ft

For anemometer readings;
Area of pipe where used, inlet, sq. ft , F_1
Area of pipe where used, outlet, sq. ft., F_2
For Pitot tube readings:
Area of pipe where used, sq. ft., F_3
For Venturi tube:
Area upstream section, sq ft , F_4
Area throat section, sq. ft., F_5
Angle of convergence, degrees
Angle of divergence, degrees
Weight of air per cubic foot under actual conditions, lbs., δ

Number of Run.	1	2	3	4	5	6
1 Speed of fan, R.P M.						
2 Velocity of fan blade tips, ft. per sec						
3 Pressure head in receiver, inches water, h_1						
4 Height of equivalent air column, ft ,						
5 Velocity head of discharge, Pitot tube, inches water, h_3.						
6 Discharge by Pitot tube, cu ft per sec , Q_3						
7. Velocity head of discharge, Venturi meter, inches water, h_5						
8. Discharge by Venturi meter, cu ft per sec., Q_5						
9 Discharge velocity by anemometer, ft. per sec V_{a_2}						
10. Discharge by anemometer, cu. ft per sec , Q_{a_2}						
11. Intake velocity by anemometer, ft per sec , V_{a_1}						
12 Intake by anemometer, cu ft per sec , Q_{a_1}						
13 Average discharge, cu. ft per sec., Q						
14 Equivalent discharge, under standard conditions, cu. ft per sec , Q'						
15. Weight discharged, lbs per sec , W						
16. Air horse power, based on average discharge.						
17. Applied horse power, A H.P						
18. Manometric efficiency						
19. Fan efficiency						
20. Ratio discharge to intake, by anemometer, $\frac{Q_{a_2}}{Q_{a_1}}$						
21. Slip and leakage, by anemometer, per cent						

CHAPTER XXIV.

MECHANICAL REFRIGERATION.

425. **Theory of Mechanical Refrigeration.** — The statement is often made that the refrigerating machine is a heat engine run backward. As a matter of fact, a refrigerating plant using a liquid refrigerant bears a much closer analogy to the steam boiler. The boiling temperature of a liquid depends upon the pressure upon the liquid. The temperature of the liquid cannot be raised beyond this point as long as any liquid remains, except by raising the pressure. Likewise, a reduction of the pressure upon a boiling liquid means a lowering of the boiling point, the difference in the heats of the liquid between the two boiling points serving to vaporize a part of the liquid. Suppose that a liquid, with a boiling temperature at atmospheric pressure considerably below average room temperature, be confined in the vessel *A*, Fig. 583. Suppose next that the liquid is kept at room temperature, so that the vapor pressure exerted by it will be higher than atmospheric. If this liquid is drawn off through pipe *P* and allowed to expand to atmos-

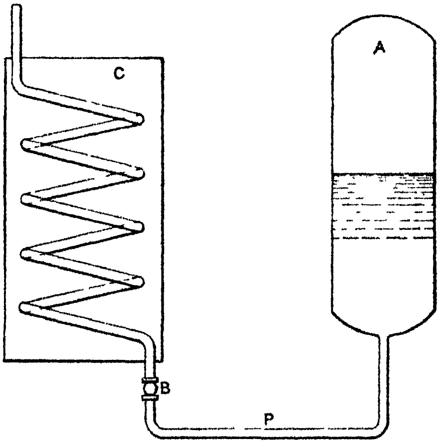

FIG. 583

pheric pressure through a suitable valve *B*, its temperature will drop, owing to partial or complete vaporization, and if allowed to circulate through coil *C*, the space in which coil *C* is placed will be cooled. This will supply a further stock of heat for vaporizing the remainder of the liquid. Theoretically, therefore, we need nothing more than a constant supply of a proper liquid (refrigerant), under a pressure higher than atmospheric, to produce a cooling effect.

There are two practical objections to this simple type of plant: first, the refrigerant used may, in the case of ammonia and sulphur dioxide, after vaporization be dangerous to health; and, second, the process is uneconomical. There are then two things to be done: the vaporized refrigerant must be prevented from escaping, and it must be brought back to its initial condition and returned to the vessel *A*. One method of doing this is shown in Fig. 584. The vapor is drawn into the compressor *E*, where its pressure is raised. It is then discharged into the condenser *G*, in which it is liquefied at constant pressure. From here it is then returned to the receiver *A* and used over again.

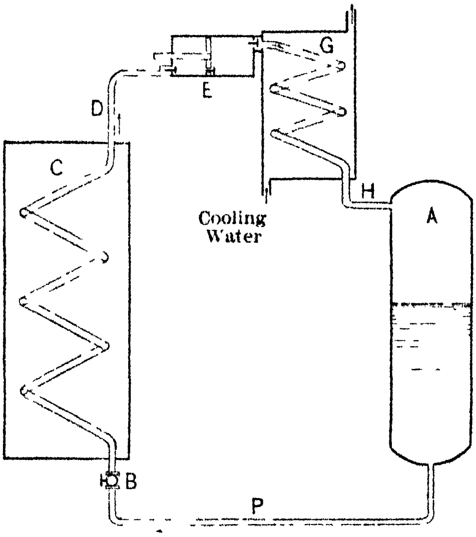

FIG. 584. — DIAGRAM OF SIMPLE VAPOR COMPRESSION REFRIGERATING MACHINE.

In its simplest form, then, a refrigerating system using a vapor may consist of a compressor *E*, pumping the vapor out of the *low-pressure side BCD* of the installation, a condenser *G* in which the high-pressure vapor is liquefied, being stored in receiver *A*. That part of the system including *E, G, H, A, P,* and *B* is known as the *high-pressure side* of the installation.

If a gaseous instead of a liquid refrigerant is used, the cycle of operations is similar, but there are two changes. In the first place, the condenser *G* merely cools the gas but does not condense it to the liquid state. In the second, if the high-pressure gas were simply allowed to expand through an ordinary expansion cock or nozzle *B*, the cooling effect would be inappreciable, because no work is done. For that reason this part of the system is replaced by an expansion cylinder in which the high-pressure gas expands behind a piston, doing work. This work may, of course, be used to help operate the compressor.

426. Refrigerating Agents. — A good refrigerating medium should have low boiling point at ordinary pressures, large latent heat of vaporization, and small specific volume. The first is desirable because it makes operation possible without the use of very high pressures in any part of the system, thus allowing the use of lighter machines, with smaller loss by leakage, etc. The latent heat of vaporization is in a sense a direct measure of the cooling effect; the greater the heat of vaporization, the better the agent. In this respect water is best, but on account of its very high specific volume it is not used. The specific volume of the agent controls the cylinder volume of the compressor per unit of refrigeration, — in other words, controls the size of the machine.

The agents that have been used or proposed are: water, air, ammonia (NH_3), carbonic acid (CO_2), sulphurous acid (SO_2), sulphuric ether, ethyl and methyl chloride, Pictet fluid, etc. Of these only air, NH_3, CO_2, and SO_2 are of any commercial importance. Pictet fluid is a combination of SO_2 and CO_2. It is seldom used. The table, p. 995, gives the main characteristics of NH_3, CO_2, and SO_2.

Various authorities give figures that are somewhat at variance with the figures of the table, particularly with reference to specific heats.

	Press. in Lbs. per Sq. In. at 0° F.	Vol. in Cu. Ft per Lb. at 0° F.	Latent Heat. r	Specific Heat of the Liquid.	Specific Heat of the Vapor. C_p	Boiling Point ° F. at Atmos. Press.	Relative Vol. of Compressor for Equal Refrigeration Effect
NH_3 .	30.0	9.00	556.0	1.05*	.53	−28.5	23.3
CO_2....	310.0	.294	117.8	.54	.217	−140.0	3.2
SO_2.	10.3	7.20	170.6	.32	.155	+15.0	61.7

* Wood, in Vol X, Trans. A. S M. E., gives for NH_3.

$t = -40°$ $C_p = 1.091$
$= +8.1$ $= 1.086$
$= +46.58$ $= 1.056$
$= +100.0$ $= 0.976$

In comparing these three agents, two sides of the question should be taken into consideration. From the practical standpoint, assuming that the limits of operation are not below −5° F., nor higher than 85°, it will be found (see tables in the Appendix) that the absolute operating pressures are: for NH_3, from 27 to 175 lbs.; for CO_2, from 290 to 1000; and for SO_2, from 9 to 65 lbs. These pressures are ordinary for both NH_3 and SO_2, but high for CO_2. The lower pressure for SO_2 is below the atmosphere and any inleakage of air may cause serious corrosion of metal by the formation of sulphuric acid. The pressures for CO_2 are so high as to cause trouble in keeping tight joints, although any leakage does no harm except for the loss of refrigerating agent. The high pressures necessary and the small specific volume make a very compact machine (see the last column in the above table), and this, combined with the fact that any leakage causes no discomfort whatever, makes CO_2 a favorite agent for use on shipboard. As between NH_3 and SO_2, the much greater latent heat of vaporization generally decides in favor of NH_3, in spite of the lower operating pressures for SO_2. Ammonia has the practical disadvantage that it corrodes brass, or any other copper alloy, very readily, and only iron can be used in the construction of those parts of the machine with which the agent comes in contact.

From the standpoint of thermal efficiency, there are again certain differences between the three agents. If the Carnot cycle is considered the standard of efficiency for the refrigerating proc-

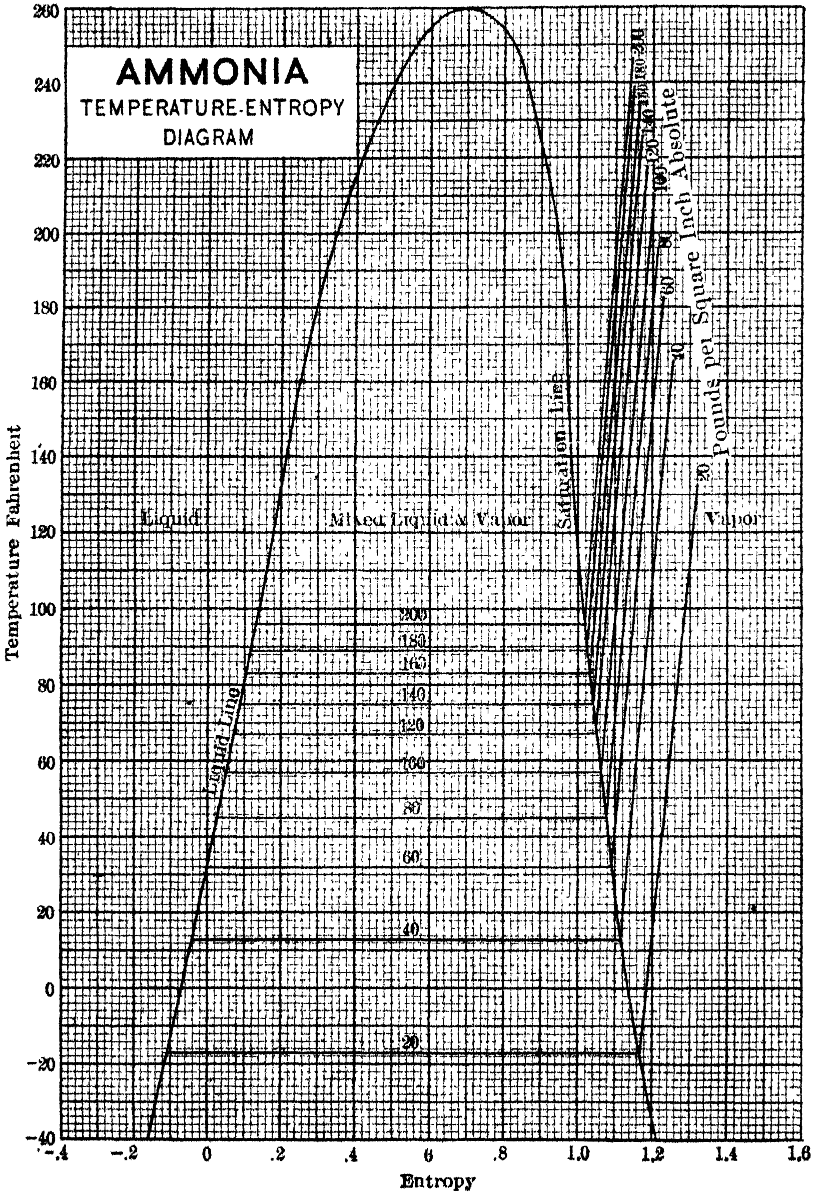
AMMONIA
TEMPERATURE-ENTROPY
DIAGRAM
Temperature Fahrenheit
260
240
220
200
180
160
140
120
100
80
60
40
20
0
-20
-40
-.4
-.2
0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
Entropy
Liquid
Mixed Liquid & Vapor
Saturation Line
Vapor
Liquid Line
Pounds per Square Inch Absolute
200
180
160
140
120
100
80
60
40
20
180-200
160
120-140
100
80
60
40
20

FIG. 585.

ess, the liquid used would be a matter of indifference, because the efficiency of this cycle depends only upon the temperature limits. The real cycle is, however, different from the Carnot, and there is a certain loss connected with this modification. It can be shown * that the loss is least for NH_3 and greatest for CO_2, and this is, of course, another factor operating in favor of NH_3.

Tables 7, 8, and 9, in the Appendix give the thermal properties for NH_3, SO_2, and CO_2 respectively. They are similar to the steam table, and the columns need no further explanation. The NH_3 table is usually given on the basis of temperature, but has in this case been recomputed to the pressure basis, following steam-table practice. The SO_2 and CO_2 tables have been left in the common forms, as these two agents are little used in this country.

Fig. 585 gives the entropy diagram for NH_3, the meaning of which should be clear from its analogy to the steam entropy diagram, Chap. XI.

The only *gaseous refrigerant* that has been used is air. Its low heat capacity and the fact that there is no change of state during the process require a much larger plant than does a liquid refrigerant for the same capacity, increasing the cost of operation. This is offset by the fact that air is cheap and has no dangerous or offensive properties. It has been used on shipboard on the latter account.

427. **Classification of Refrigerating Machines.** — The following scheme shows at a glance the various classes of refrigerating machines now in use.

I. Machines using air.	1. Machines in which the air is used over and over again without coming actually in contact with substance to be cooled, called the closed-cycle machine. 2. Those in which the cold air is circulated through the rooms to be cooled, called the open-cycle machine.
II. Machines using an agent that is alternately condensed and vaporized.	1. Machines which use heat directly to produce cold. Absorption machines and vacuum machines. 2. Those which produce cold by the expenditure of mechanical energy. Compression machines.

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* See Ewing, The Mechanical Production of Cold, p. 66.

428. Air Machines. — The principle upon which these machines operate is very simple. In Fig. 586, if air at 60° F. and atmospheric pressure is taken into the compressor *A* and compressed adiabatically to 60 lbs. per square inch, the temperature will be raised to 320° F. This air is then transferred to the water-cooled condenser *C*, cooled to 60° F., and is then allowed to do work in the expansion cylinder *B*, expanding to atmospheric pressure. In going through this cycle, the theoretical end temperature should be − 113° F. Practical losses of course modify this theoretical result. The expansion cylinder *B* is connected to the same shaft as the compressor cylinder *A*, so that the work of expansion is utilized to help compress the air. The deficiency produced by friction and leakage losses is made up by some source of power *D*. If the air, after expanding in *B*, is next circulated through a system of cooling pipes, in which case expansion in *B* need not be down to atmosphere, we have Class 1 of air machines,— the closed-cycle machine. The air, after doing its work, forms the suction supply for compressor *A*. In the open-cycle machine, the cold air is forced directly into the rooms to be cooled and is continuously displaced

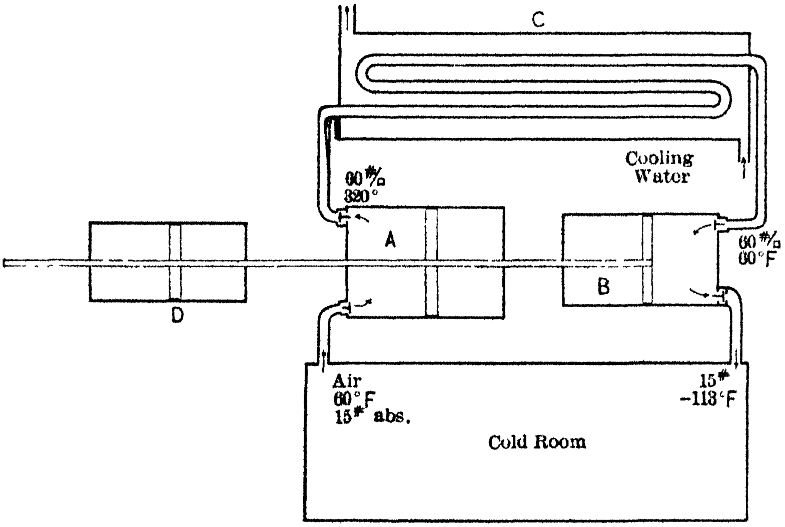

FIG. 586.—DIAGRAM OF REFRIGERATING MACHINE USING AIR, CLOSED CYCLE.

by a new supply. The latter type of machine is to-day more used than the closed-cycle machine.

Cold-air machines have the advantage of great simplicity, and, neglecting efficiency, are satisfactory for small capacities. For large capacities, the capital cost of the installation makes the operation no longer profitable in competition with other machines. The difficulty of moisture freezing in the expansion cylinder and choking up ports and valves can be overcome by the interposition of a drier between the cooler and the expansion cylinder, so that smooth operation is possible. The efficiency of operation is low as compared with other machines.

429. The Absorption System. — This will be described with ammonia as the refrigerating agent. An absorption plant consists of the following essential parts: the generator, the condenser, the refrigerator, the absorber, the interchanger, exchanger, or economizer, the rectifier, and the analyzer. The interrelation of all these parts is shown in the conventional sketch, Fig. 587. The operation of the plant depends upon the fact that water will absorb NH_3 gas, the quantity depending upon pressure and temperature conditions. Table 10, Appendix, gives some figures for the absorbing power of water for NH_3, from which it will be seen that at any one temperature the absorbing power increases with the pressure, while at any one pressure the absorbing power decreases as the temperature increases. The latter fact is made use of in the absorption machine to obtain NH_3 vapor under pressure, and the reverse of this action is used to make the vapor combine again with water after it has done its work.

To explain the operation from the sketch of Fig. 587, the generator contains a quantity of water holding a large quantity of NH_3 gas in solution (strong liquor). The temperature of this liquor is raised by admitting steam to the heating coils shown, the condensate being removed by trap or other means. The heating starts to drive the NH_3 gas out of the liquor, but, since the system is a closed one, the vapor pressure above the liquor will also be raised until equilibrium is established, depending upon temperature, pressure, and initial strength of liquor. The NH_3 gas liberated, which carries with it a certain small quantity of water vapor, rises and passes through a vessel called the analyzer. Here it meets with a descending rain of comparatively cool, strong liquor, which is on its way to the generator. Fine subdivision is produced by having the incoming strong liquor run down over a number of perforated trays. The result is that the descending liquor takes a certain quantity of heat from the ascending NH_3 gas. This is desirable, because the liquor must be heated in the generator, and the gas must be cooled before it can be used. Hence the analyzer acts as an economizer. The wet gas rising through the analyzer next passes into a rectifier, which is practically a condenser cooled by water circulation. Here the gas is cooled, not enough to liquefy, but sufficient to condense the water vapor carried by it, which is returned to the analyzer by a drip pipe. The NH_3, therefore, leaves the rectifier in a dry state. The gas is next passed into a water-cooled condenser, where liquefaction is produced by the cooling. The liquid NH_3 is stored in the receiver and is from here

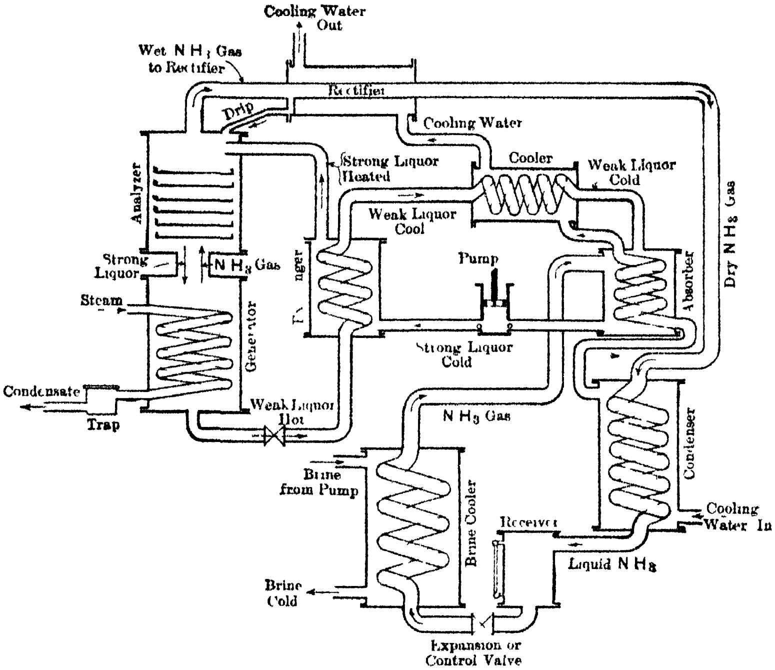

FIG. 587. — PARTS OF AN AMMONIA ABSORPTION SYSTEM

passed through a control or expansion valve into the cooler. A brine cooler is here indicated. The liquid NH_3 partially gasifies on passing the valve, the rest of the gasification being finished in the brine cooler. The NH_3 gas leaving the cooler then passes into the absorber, in which it meets and is mixed with cold, weak liquor, by which it is absorbed. This produces strong liquor, which is forced by the pump through the exchanger into the analyzer, and so into the generator. This closes the cycle.

In the generator, weak liquor works toward the bottom on account of its greater specific gravity. It is passed through coils in the exchanger, in which it is cooled by contact with the cold, strong liquor on its way to the analyzer. This again is a desirable heat interchange, because the strong liquor must be heated in the generator and the weak liquor, which is hot coming from the generator, must be cooled before entering the absorber. The exchanger, like the analyzer, is therefore simply an economizer. The cooling of the weak liquor is generally completed in a separate cooler between the exchanger and absorber.

It must have been noticed that all possible precautions have been taken in this system to economize heat. The same may be said with regard to the consumption of cooling water. This does service first in the condenser and then cools in order the absorber, the weak-liquor cooler, and the rectifier.

That part of the plant including the generator, the analyzer, the rectifier, the condenser, the receiver, and the piping from one side of the liquor pump to the generator and from the receiver to the control valve, is the high-pressure side of the plant. The low-pressure side includes the brine cooler, from the expansion or control valve, the absorber, and the piping up to the pump.

Fig. 588 gives a better idea of the actual relative size and construction of the various parts of an absorption plant. In this installation the rectifier is combined with the condenser, being located above it.

430. The Vacuum Process. — This process employs water as the refrigerating agent. It is one type of absorption process. The principle of operation is quite simple.

In Fig. 589, let the water to be cooled or frozen be placed in the

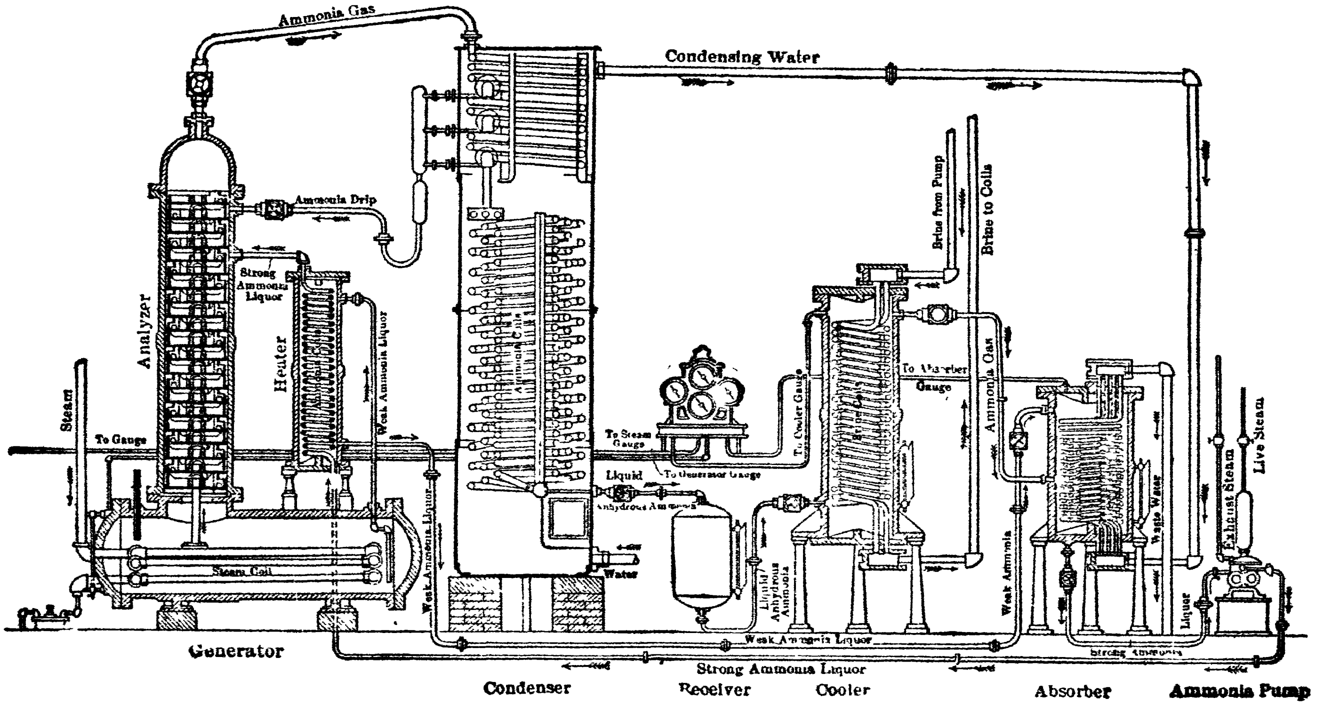

FIG. 588.—AMMONIA ABSORPTION PLANT.

vessel *B*, and let this vessel be connected to a vacuum pump *A*, by means of which the pressure above the water is lowered to below the vapor pressure. The water then begins to boil, taking its heat of vaporization from the water itself and lowering the temperature. The water vapor fills the space *E* over the water and the space *D* in an absorber, which is filled with strong sulphuric acid. Arrangements are made to promote thorough mixing between the vapor and the acid to cause rapid absorption. The pump *A* handles practically no vapor and serves merely to maintain a high vacuum. The cold water is either circulated through rooms to be cooled, or the operation may be carried on until the water freezes. The ice made is rather "mushy" and must be compressed.

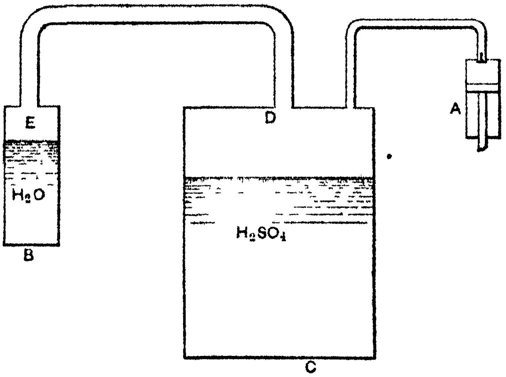

Fig. 589.—Principle of Vacuum Process.

The rest of a commercial vacuum plant consists of arrangements for continuous concentration of the acid, which of course is constantly diluted in the absorber during the operation of the plant.

The efficiency of the vacuum process is probably as high as that of any of the other methods discussed, but in practice it is found that the strongly corrosive action of the acid fumes makes it hard to keep the system tight, which is of course a necessity on account of the high vacuum that must be maintained for best efficiency.

431. The Vapor-compression System. — The principle of operation of this system has already been outlined in Art. 425 and Fig. 584. Except for the difference introduced in operating pressures, temperatures, compressor sizes, etc , pointed out in Art. 426, vapor-compression machines are very similar whether they employ CO_2, SO_2, or NH_3. Since the first two agents are not used in this country to any extent, the discussion will be confined to the ammonia machine.

A common arrangement of compression plant is shown in Fig. 590.* The compressor used is of the double-acting steam-driven

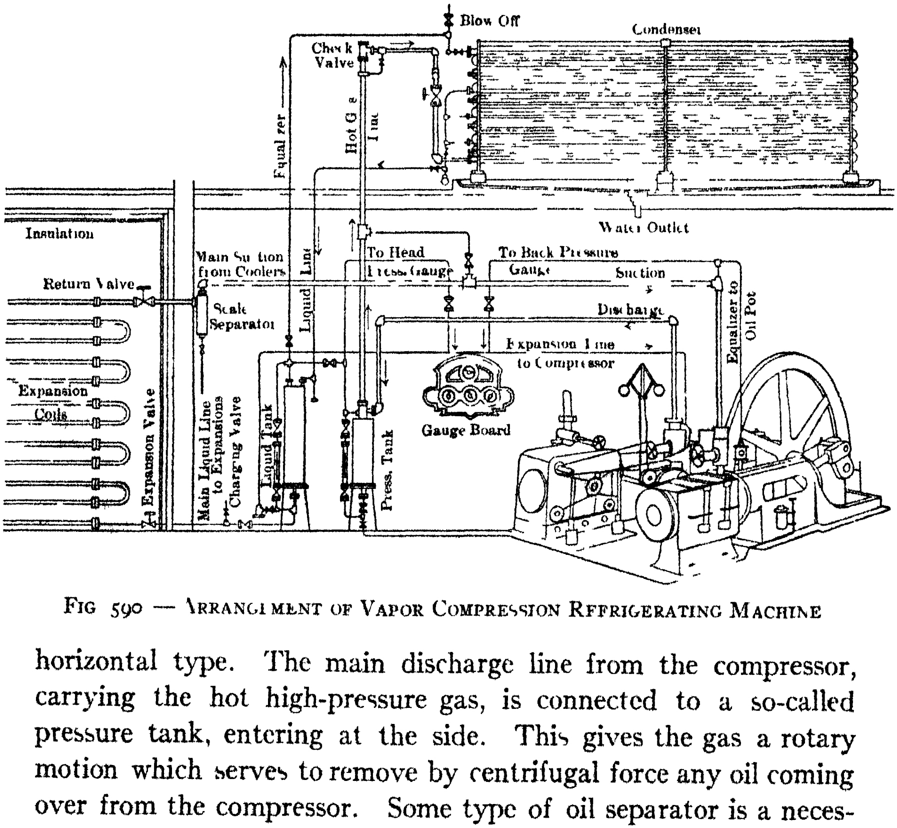

Fig 590 — Arrangement of Vapor Compression Refrigerating Machine

horizontal type. The main discharge line from the compressor, carrying the hot high-pressure gas, is connected to a so-called pressure tank, entering at the side. This gives the gas a rotary motion which serves to remove by centrifugal force any oil coming over from the compressor. Some type of oil separator is a necessity to prevent the coating by oil of the condenser coils, which would seriously decrease the efficiency of heat transfer. The gas then passes through the hot-gas line, which makes a loop, and enters a header connected with the bottom coils of the condenser. The purpose of the loop is to prevent liquid NH_3 from running down into the pressure tank if the compressor should be shut down. The type of condenser here used is the atmospheric, the hot gas passing from the bottom coils toward the top, while water is trickled down from above over the outside of the coils. As the NH_3 gas liquefies, it is carried out of the coils by smaller drip pipes connected to a common liquid header. The pipe from this header should rise in a small loop, to keep the header always full of liquid and to prevent gas from entering the liquid line. The liquid line leads down to the liquid tank. If gas should come down this line, it may be led back to the condenser by means of the equalizer line shown. From the liquid tank or receiver the liquid NH_3 next flows to the expansion valve. This in most cases is nothing but a valve, the opening of which can be readily controlled to any desired size. The pressure on the liquid, in passing this valve, is suddenly reduced from 150-200 lbs. to say 20 lbs. per square inch. This of course causes immediately a partial vaporization, but, since it takes time to supply to each pound of liquid NH_3 the total latent heat of vaporization required (about 550 B.t.u. in the ordinary case), the vaporization is by no means instantaneously completed, as is often supposed. The vaporization is completed in the expansion coils, the heat in this case coming from a room to be cooled. The vapor is returned through a return valve and through the suction line to the compressor, passing on its way a scale trap intended to prevent any scale forming in the pipe system from being carried over into the compressor. Between the liquid receiver and the expansion valve there is usually placed a side branch supplied with a valve. This serves as a charging connection, the ammonia drum containing liquid NH_3 being connected at this point and as much ammonia being allowed to flow into the line as the system requires, as shown by control gauges on the gauge board. There is a small pipe line running from the liquid receiver to the suction side of the compressor (in the figure called "Expansion Line to the Compressor"). This line is furnished with an expansion valve through which cold gas may be supplied to prevent undue heating of compressor on starting or when the return gas is not cold enough for satisfactory operation. The compressor is also often furnished with by-passes and pump-out lines, by means of which the compressor may reverse its operation, taking NH_3

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* Reproduced from an article by F. E. Matthews in *Power*, Sept. 29, 1910.

from the condenser and forcing it into the expansion coils, for the purpose of pumping out parts of the system when repairs become necessary.

Commercial plants show numerous modifications in construction, but not in principle, from the features above described.

Compressors are built *single-* or *double-acting*, *horizontal* and *vertical*. A favorite combination is a pair of single-acting vertical compressor cylinders operated by a horizontal Corliss engine connected to the same shaft, as in Fig. 591. The tendency to-day is toward single-acting compressors with the suction valve located in the piston and with poppet-lift discharge valves in the head. Some builders construct the head so that it may be bodily lifted by the piston against the resistance of springs (see Fig. 592). In single-acting compressors, the suction end of the cylinder is always

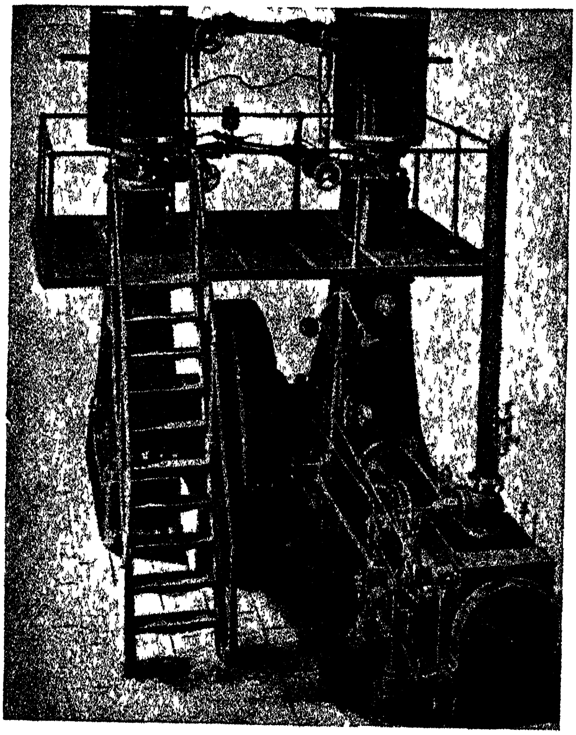

FIG 591 —VAPOR COMPRESSION REFRIGERATING MACHINE

under suction pressure only. The stuffing box, through which the piston rod passes, is therefore called upon to hold tight only against this pressure, and leakage at this point is materially reduced. One of the most important features of compressor-cylinder design is the attainment of the smallest possible clearance, to decrease the amount of gas remaining at the end of the stroke. If this matter is not taken care of, there will be a large amount of re-expansion with consequent loss of compressor capacity (low volumetric efficiency).

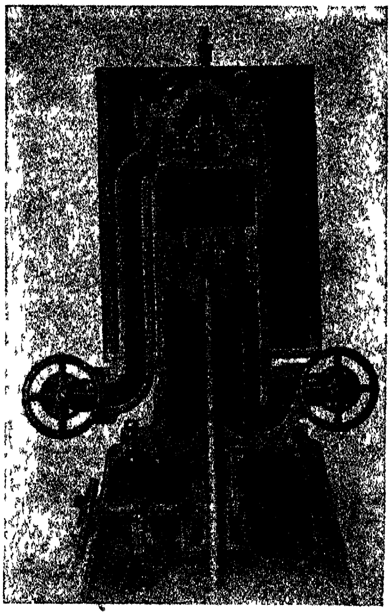

FIG. 592.—CYLINDER OF COMPRESSION REFRIGERATING MACHINE, SHOWING ONE METHOD OF ELIMINATING CLEARANCE.

Condensers may be of three types: the *submerged coil*, the *atmospheric*, and the *double* or *concentric*. In the first the ordinary coil is placed in a tank of water, the water being continuously circulated. Where water is expensive, the atmospheric type or the concentric type is used. The former of these is shown in Fig 590. In this condenser the use of the counter-current principle, in connection with the fact that a considerable part of the finely divided water is evaporated (the heat for the latter process coming from the NH_3), makes this type more efficient than the submerged coil type. The most efficient type is probably the double or concentric condenser. In this condenser each pipe element consists of an outer and an inner tube. The cooling water is circulated through the inner tube, while the NH_3 fills the annular space. Thus cooling is done both by water on the inner surface and by air surrounding the outer pipe.

The *cooling system* may be the *direct* or the *indirect* or *brine system*. The former is shown in Fig. 590, the cold gas being circulated through pipes in the room to be used. If, however, an ammonia leak would do damage in the cold-storage room, the indirect system is used, in which some other liquid, usually brine on account of its low freezing point, is cooled by the ammonia and circulated for cooling purposes instead of the ammonia. One type of brine cooler is the submerged coil cooler, where the ammonia-expansion coils are immersed in a brine tank, the cold brine being pumped from the bottom, circulated, and returned warm at the top. The concentric pipe cooler is rather more convenient, however, and is very largely used. The brine flows in the inside pipe, while the expanding ammonia is in the outer one. The principle of counter-current flow is used in the concentric pipe to get the maximum possible heat exchange.

432. **Ice Making.** — There are three systems of ice making in use: the *can*, the *plate*, and the *cell system*. Only the first two, however, are of any importance in this country. In the can system, tapered galvanized-iron cans of the size of ice cake desired are filled with water and placed in a brine tank through which pass the pipes circulating the cold brine from the cooler. The temperature of the brine in the tank is usually kept at about 14° F. The time required for freezing depends of course upon the size of the cake, a can 8″ × 8″ × 31″ requiring about 20 hours' immersion, forming 50 lbs. of ice, while one 11″ × 22″ × 44″ requires about 60 hours, the ice weighing about 400 lbs. After the water is frozen, the cans are hoisted out of the tank, and the ice is loosened by steam. The economy of this system depends of course upon conditions of operation and grade of machinery used. With a head pressure of 190 lbs., a suction pressure of 15 lbs., and condensing water at 70°, Professor Denton estimates that the average-economy figures should be about 6 lbs. of ice per pound of coal used.

In the plate system, plates through which the freezing solution circulates are immersed in a tank of water. The water next to the surface of the plate freezes, and the thickness of the ice increases with time. After the desired thickness is reached the plate, with the ice attached, is lifted out of the tank, the ice being thawed off by passing hot gas or brine through the plate coils. It is not common to place the plates so close together that the ice formed on the adjacent plates will meet, as the core so formed is usually ice of poor quality. Good practice in the use of the plate system should show an economy of from 10 to 12 lbs. of ice per pound of coal.

The cell system is a modification of the plate system. The water is placed in cells constructed of hollow walls through which the freezing solution is circulated. This system is not in use in this country.

As far as the quality of the water required for ice making is concerned, it is necessary, unless distilled water is used, to agitate the water during freezing to free it from air, if clear ice is desired. In any case, of course the water should be as pure as possible, and hence the extended use of distilled water. The preference shown for clear ice is largely sentimental; for the opaque ice, which results if the water is neither agitated nor distilled, is fully as satisfactory for most ordinary cooling purposes as clear ice, provided the water is pure, and is certainly cheaper to make.

433. **Properties of Brine Used for Cooling.** — Two kinds of brine are in use, — calcium-chloride and sodium-chloride brine. The addition of either chemical to water lowers the freezing point of the latter. Naturally the brine is made of such strength that its freezing point is somewhat below the lowest temperatures of operation. The brine tables in the Appendix* show that for the same strength of brine, as measured by the per cent by weight of the chemical contained in the water, $CaCl_2$ brine shows a much lower freezing point than $NaCl$ brine. It is therefore easier to obtain a low freezing point brine by the use of $CaCl_2$. Sodium brine, if very strong, shows a tendency to deposit salt in the coils, which may seriously interfere with operation. It also strongly rusts any iron parts with which it comes in contact, a disadvantage not possessed by calcium brine. The specific heat of calcium brine is higher than that of sodium brine of the same strength, which is another advantage, as less of the brine needs to be circulated. Thus, although calcium brine costs somewhat more than sodium brine, the former is generally used and preferred. A 20 per cent solution is commonly employed.

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* Taken from H. Williams' "Mechanical Refrigeration."

The strength of brine is best expressed on the ordinary specific-gravity scale, using a hydrometer. In practice, an instrument called a *salometer* or *salinometer* is often employed for measuring strength. This instrument reads 0 in clear water and the strength of brine is indicated by a reading in degrees between 0 and 100. It is stated by some authorities that the 100° mark corresponds to the saturated solution, but existing brine tables do not quite agree on this; hence the statement above, — to use specific gravity to avoid errors.

434. Insulation. — With the extended application of cold-storage work, the proper insulation of cooling rooms has become one of the most important questions of refrigeration practice. It must be evident that, once products have been cooled to the temperature desired, it is merely necessary to supply enough refrigeration to make up the heat gain from without to maintain that temperature. The heat gain is less, the greater the efficiency of the insulating material.

The materials used are hair felt, mineral wool, different varieties of cork products, rock wool, wood shavings, etc., used in combination with wood, cement, masonry, and air spaces. Dead air is a good nonconductor, but a simple air space, unless of exactly the proper width, may have set up in it down-currents on the cold surface and up-currents on the outside warm surface, favoring convection. It is therefore the common practice to break up the air space by filling it with some porous material, to form an infinite number of small dead-air spaces, effectually preventing the setting up of any air circulation. Too small air spaces, resulting from material too tightly packed, may, however, again favor the conduction of heat across. Insulating material should be moisture-proof, if possible, and at least slow-burning. For the method of constructing nonconductive walls, see any good book on Refrigeration.*

If a block of any material is exposed to a certain temperature on one side and to another lower temperature on the other side, there will be a transfer of heat from the former to the latter side, which transfer is a function of the temperature difference of the kind of material and of the thickness used. The number of heat units that will be so transferred per degree difference of temperature per hour and for 1 in. in thickness is called the *coefficient of the material.* It should be noted that this heat loss is also largely a function of the velocity of air on both faces of the material. The constants given in the following table for some of the well-known material holds for still air only. They increase if there are air currents over either face.

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* Lorenz-Pope-Haven-Dean, Modern Refrigerating Machinery, gives a number of examples. A good discussion is contained also in "Refrigeration, Cold Storage, and Ice Making," by Wallis-Taylor. See also Göttsche, Die Kältemaschinen und ihre Anlagen.

COEFFICIENT OF HEAT TRANSMISSION

1 Sq. Ft, 1″ Thickness, 1° F. Difference of Temp, 1 Hour.

Material	Coefficient Btu	Material	Coefficient Btu
Air, still	30	Corks..	1 0 1 8
Asbestos	1 20	Paper	28
Cotton	28	Sand .	2 1-3 0
Glass	3 5-5 2	Shavings	•45- 65
Hair felt	50-2 1	Mineral wool ..	5 7
Wood, average	1 10	Masonry, brick ...	5 0
Wood ashes	45	Masonry, stone	3 5
Kieselgur (infusorial earth)	50	Coal ashes	7
Coke, pulverized	1 20		

There are several methods of *testing insulation.* One of the most common is to erect a length of steam piping on a slight slant and to furnish to the higher end of the pipe steam under any given pressure and quality. At the lower end a collector for the condensed steam is arranged. This may be simply constructed of a vertical length of pipe furnished with a gauge glass and fittings at the side. The steam condensing flows into the collector by gravity and is drawn off and weighed from time to time, a nearly constant level being maintained in the collector. Since the condensate drawn off is under the pressure of the steam, and would consequently largely flash into steam if this pressure is suddenly released, it is necessary to draw off into cold water. The quantity of heat radiated from the pipe in a given time is then computed from the weight of the condensate in the same time multiplied by the quality of the steam and by the heat of vaporization of the steam for the given pressure. The other important item in this test is the temperature difference. This should strictly be taken as the difference between the temperatures of the inner and the outer wall skin and not of the usually gaseous media adjacent to these walls. In this case the temperature of the steam under the given pressure may be fairly taken as the temperature of the inner wall. To get that of the outer wall would require the use of a resistance thermometer. Instead of this, the common practice is to suspend thermometers at several points in the air close to the pipe but not in contact, and to assume the temperature difference equal to the steam temperature minus the average air temperature so found.

The common practice is to first test the pipe bare and then again when covered with the insulation to be tested. In each case the quantity of heat transmitted per square foot of pipe surface per degree difference of temperature per hour is computed. If Q_b represents this quantity for the bare pipe and Q_c the quantity for the covered pipe, then the *efficiency of the insulation* is

$$E = \frac{Q_b - Q_c}{Q_b}.$$

Note that Q_c is not the coefficient spoken of above, because the definition of this coefficient requires that the insulation be 1 in. thick. It should be stated that it will not do to divide Q_c by the thickness of the insulation to obtain the coefficient, for the reason that the efficiency of insulation does not vary directly with the thickness.

Referring more specifically to the kind of insulation used to protect cooling compartments, another method of testing would be to determine the electrical energy input to keep a compartment at a certain temperature with a given temperature outside of the compartment. Depending upon circumstances, a number of other methods will suggest themselves.

435. The Ideal Vapor-compression Refrigerating Cycle. — The ideal cycle for a vapor-compression machine is the Carnot. Starting with the liquid NH_3 and leaving the condenser at an absolute temperature T_1, we locate point A on the liquid line of the entropy diagram, Fig. 593. This liquid next enters the expansion cylinder, vaporization taking place in this cylinder until the lower absolute temperature T_2 is reached, locating point B. The vaporization is only in part completed, the quality of the mixture after leaving the expansion cylinder being represented by the ratio $\frac{B'B}{B'C}$. Vaporization is continued in the refrigerator (really the heater as far as the refrigerating agent is concerned). At the end of the refrigerator action and by the time the compressor is reached, the vapor may be wet saturated, as indicated by any point between B and C, dry saturated at C, or superheated as indicated at C''. What the end condition of the vapor is depends largely upon the amount of refrigerating medium in circulation. At C' the quality is such that the ensuing adiabatic compression $C'D'$ (wet compression) to the temperature T_1 just renders the vapor saturated. This is indicated by D'. After compression the hot high-pressure vapor is then sent to the condenser, in which it is assumed that

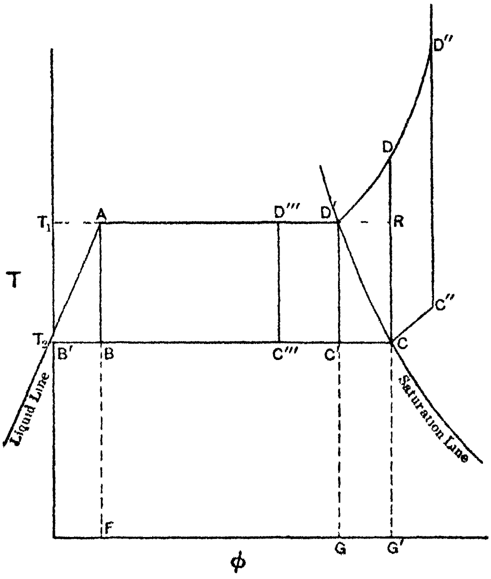

FIG. 593.—IDEAL VAPOR COMPRESSION REFRIGERATING CYCLE.

heat is abstracted to the extent that liquefaction is just completed. This process is indicated by line $D'A$, which completes the cycle.

If the refrigerator action is stopped so that the vapor quality is even less than $\frac{B'C'}{B'C}$, the complete ideal cycle, if carried out as before, will be represented by the area $ABC'''D'''A$. If vaporization is continued to dryness at C, the ensuing adiabatic compression will superheat the vapor, the process being shown by line CD (dry compression). The superheated vapor is then cooled in the condenser, first at constant pressure along DD', and then liquefied along $D'A$. Finally, if the action of the refrigerator is such as to superheat the vapor at the inlet to the compressor, the compression will proceed along a line $C''D''$, and the complete ideal cycle is the area $ABCC''D''D'A$.*

436. The Ideal Coefficient of Performance. — It will be noted by inspection that the ideal cycle outlined in the previous article is an ideal heat-engine cycle (Carnot) reversed. In the heat-engine cycle, a certain quantity of heat Q_1, represented by the rectangle AG in Fig. 593, is taken in along the line AD'. A certain other quantity of heat Q_2, represented by the rectangle BG, is discharged along the line $C'B$. The external work W done is represented by the rectangle AC', and we have of course

$$W = Q_1 - Q_2. \tag{1}$$

The same reasoning may also be applied to the other cases indicated in Fig. 593.

In the refrigerating cycle, we remove from the body to be cooled the heat Q_2, equivalent to rectangle BG; in the compressor we do a certain amount of work W, equivalent to rectangle AC', and discharge a certain amount of heat Q_1, equivalent to rectangle AG, into the condenser.

If we define efficiency as the ratio of the useful effect of an operation to the effort or energy expended to gain this effect, we evidently will have for the heat engine

$$\text{Efficiency} = \frac{W}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1}, \tag{2}$$

---

* $ABCDD'A$ and $ABCC''D''D'A$ are not Carnot cycles.

and for the refrigerating machine

$$\text{Efficiency} = \frac{Q_2}{W} = \frac{Q_2}{Q_1 - Q_2} = \frac{T_2}{T_1 - T_2}. \tag{3}$$

The latter expression is nearly always greater than unity, and can in a certain sense only be looked upon as an "efficiency." What the factor really expresses is the number of times the heat W expended in compressor work is regained in cooling effect, and the term "*ideal coefficient of performance*" instead of "efficiency" is therefore commonly used for the ratio $\frac{Q_2}{W}$.

The last form of the expression for the ideal coefficient of performance in Eq. 3 will serve to point out one or two important facts with reference to efficiency of operation. T_2 is the temperature of the cooler, while T_1 is that of the condenser. By inspection of the expression we note the following: (1) For any given T_2, the lower T_1 the higher will be the coefficient of performance; (2) for any given T_1, the higher T_2 the higher the coefficient; (3) in general, the nearer together T_1 and T_2, the better the performance. The entropy diagram also shows these facts very clearly. Thus in Fig. 594, let the condenser temperature be fixed at $T_1 = 460 + 70 = 530°$ F. abs., and let the temperature of the cold body be in case (*a*) $- 10°$ ($= 450°$ abs.); case (*b*), 0° ($= 460°$ abs.); and case (*c*), 20° ($= 480°$ abs.). For the same amount of energy W expended (shaded areas equal), the heat Q_2 removed is in case (*a*) equal to rectangle AB, in case (*b*) to rectangle $A'B$, and in case (*c*) to rectangle $A''B$. The ideal coefficients of performance are 5.6, 6.6, and 9.6 respectively. Or, conversely, for the same quantity Q_2 of heat to be removed, it requires a greater expenditure of energy W to do this between $- 10°$ and 70° than between 20° and 70°, as will be clear from a study of an entropy diagram applying to the two cases. Finally, the ideal coefficient

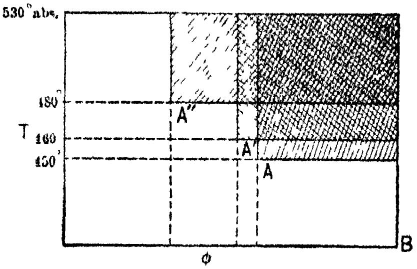

FIG. 594.

of performance depends only upon absolute temperature and is therefore independent of the working medium used.

437. The Actual Vapor-compression Refrigerating Cycle.—In an actual refrigerating machine the cycle of operation fails to meet the conditions of complete reversibility, as outlined in Art. 436, and for that reason the actual coefficient of performance is less than in the ideal case. To see the causes for this, it will be instructive to follow the refrigerating medium around an actual cycle. The study is best made on the basis of entropy diagram, and, since in the majority of cases in actual operation the vapor is superheated at the suction valve to the compressor, this will be the only case considered in detail.

Starting with the liquid in the condition defined by the point *A* on the entropy diagram, Fig. 595, this liquid is next allowed to pass through the expansion valve, which results in a lowering of pressure. We have in this process the first deviation from the ideal cycle, since an expansion valve is substituted for the expansion cylinder. The expansion is largely unresisted, the process is no

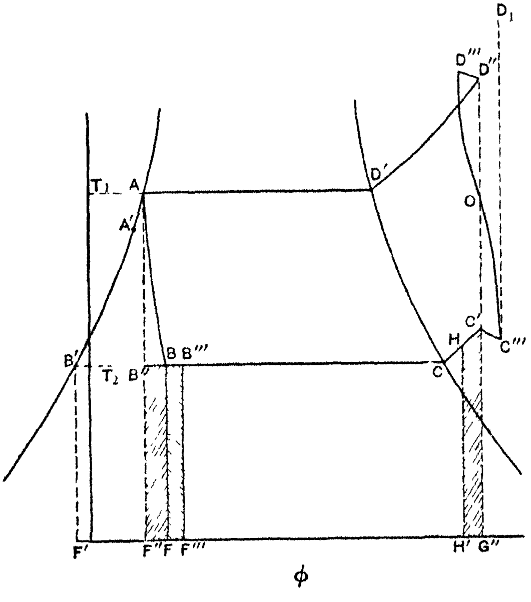

FIG. 595.—ACTUAL VAPOR COMPRESSION REFRIGERATING CYCLE.

longer adiabatic, and is irreversible. The reduction of the pressure from that existing at A to that corresponding to the temperature T_2 results in a partial vaporization of the liquid as it passes the valve. The problem is to locate the point B. If the temperature T_2 is measured at a certain distance beyond the expansion valve, so that the kinetic energy changes that accompany the passing of the liquid through the valve have again disappeared, it may be fairly assumed, neglecting radiation, that the process up to this point is one of constant heat. This means that the area $AF''F'B'A$ is equal to the area $B'BFF'B'$, which at once locates the point B, since the first area and the distance $B'F'$ are known. Mathematically, the problem may also be solved as follows: The total heat content in the vapor at A is assumed the same as the total heat content at B. Hence

$$x_A r_A + q_A = x_B r_B + q_B. \qquad (4)$$

But at A, $x = 0$, therefore, solving for the quality of the mixture at B,

$$x_B = \frac{q_A - q_B}{r_B}. \qquad (5)$$

x_B is the ratio $\frac{B'B}{B'C}$ in the diagram. The rest of the vaporization is carried on in the cooler as before, except that there is some loss of effective cooling due to the heat influx in the piping leading to and from the cooler. It is assumed that the vapor is superheated to C'', as determined by a thermometer in the suction inlet to the compressor.

For the fixing of point C'' we have the following data: Entropy at point C can be found directly from the NH_3 vapor table according to the pressure, or temperature T_2. The temperature at C'' is found directly by thermometer in the suction pipe just ahead of the compressor; call it T_3. The added entropy due to superheat is then $= C_D \log_e \frac{T_3}{T_2}$, in which C_D is the mean specific heat of the superheated vapor between T_2 and T_3. The uncertainty of the computations lies in the fact that very little is known about the value of C_D. The data available all apparently apply to atmospheric pressure. Landoldt and Börnstein give the following:

Temp. range, deg. C.	23–100	27–200
C_D.	.5202	.5356

It is probable that C_D varies for NH_3, with temperature and pressure, in a similar manner as for steam, but it is also probable that the changes are not as pronounced. At any rate, in the present state of our knowledge, the best that can be done is to assume C_D constant and equal to 0.53.

In passing the inlet valve to the compressor there is a certain loss of pressure due to wire drawing. The extent of this can be found by comparing the suction pressures just ahead of the inlet valve with those on the compressor-indicator diagrams. Considering this a constant-heat change, the process can be indicated on the entropy diagram by the line $C''C'''$. It should be understood that this change is so small ordinarily as to be negligible; it is much exaggerated in Fig. 595.

To construct the compression line $C'''D'''$ in the entropy diagram, it will be necessary to turn to the indicator card from the compressor. The method of computing temperatures along the compression line of the indicator card will be outlined in Art. 440. Assuming that for any given pressure the saturation temperature is T_s and the temperature on the compression line is T_v, the total entropy of the superheated vapor at the pressure chosen and the temperature T_v will be

$$\phi_{\text{total}} = \phi_{\text{sat. at } T_s} + C_D \log_e \frac{T_v}{T_s}. \qquad (6)$$

By aid of this equation a number of points may be located to determine line $C'''D'''$ in the entropy diagram.

At D''', the vapor is next forced through the discharge valve, the action being accompanied by a pressure drop, which may be represented by the constant-heat change $D'''D''$. In the condenser side of the system, the vapor is next cooled at constant pressure along line $D''D'$ to saturation at D'; the latent heat is removed along $D'A$. If the condenser action continues beyond that, supercooling the liquid, this action may be represented by a line AA'. In that case the action in the expansion valve will first be to retrace line $A'A$, and the cycle then starts from A, as before.

In the analysis above made, the actions along $C''C'''$ and along $D'''D''$ are much exaggerated. The compression line $C'''D'''$ slopes generally to the left, as shown, on account of heat loss to the jacket of the compressor. The amount of the deviation from the adiabatic $C'''D_1$ is of course a function of the effectiveness of cooling. It will in general be found that the points C'' and D'' are in a vertical line, or very nearly so, and that the area $OD'''D''$ is so nearly equal to area $OC''C'''$ that the entire analysis may be shortened by simply drawing in the line $C''D''$. The effect of this simplification upon the area of the entropy diagram is generally negligible.

The entropy analysis for the case of vapor wet at the suction valve is very difficult to carry out because of the practical impossibility of determining the quality of the vapor at that point.

438. The Actual Coefficient of Performance and the Real Efficiency of the Actual Refrigeration Process. — The total refrigeration effect shown by the entropy diagram, Fig. 595, assuming that an expansion cylinder is used, is equivalent to the area $B''CC''G''F''$. In the real case, this effect is reduced by the following items:

(*a*) The loss due to the substitution of an expansion valve for the expansion cylinder. This is measured by the area $B''BFF''$.

(*b*) The useless refrigeration in the piping leading from the expansion valve to the cooler. This may be represented by an area $BB'''F'''F$. In any practical case this can probably not be directly determined, for its effect would simply be to change the quality of the vapor, and we have no direct means of determining the latter.

(*c*) The useless refrigeration in the piping leading from the cooler to the suction valve of the compressor. This may again be represented by an area $HC''G''H'$, and is susceptible of direct measurement if the vapor leaves the cooler superheated.

The net refrigeration shown by the diagram is then represented by the area $B'''CHH'F'''B'''$. Call this amount Q_{net}. The work W expended is in this case equal to the area $ABCC''D''D'A$, and, by definition, the coefficient of performance as per entropy diagram is then $= \frac{Q_{\text{net}}}{W}$.

Now the quantity Q_{net} is not the useful refrigerating effect. It would be if in the cooler all of the heat given to the NH_3 came from useful refrigeration. In practically every case, however, there are radiation and conduction of heat serving no useful purpose. Let the actual useful refrigeration be represented by Q_3, and the heat added through radiation or conduction in the cooler by Q_4, then

$$Q_{\text{net}} = Q_3 + Q_4. \qquad (7)$$

Q_3 may be directly determined. In case brine is used, for instan

$$Q_3 = GC_p(t_1 - t_2) \text{ B.t.u.},$$

in which G = weight of brine in circulation per pound of NH_3;
C_p = mean specific heat of brine;
t_1 = temperature of brine entering cooler;
t_2 = temperature of brine leaving cooler.

The compressor work W is the same as before, and the *actual coefficient of performance* is therefore $\frac{Q_3}{W}$. The ratio $\frac{Q_3}{Q_{\text{net}}}$ may be regarded as the *efficiency of the cooler*.

In practice the actual coefficient of performance is computed from the useful refrigeration effect and the heat equivalent of the compressor work in a given time.

If we define the *real efficiency of the refrigeration process* as the ratio of the actual useful refrigerating effect to that theoretically obtainable for any given work W in the compressor, we will have in this case

$$\text{Real efficiency} = \frac{Q_3}{\text{Equivalent of area } B''CC''G''F''}. \qquad (9)$$

In case the vapor is not superheated at any stage of the process, the ideal coefficient of performance is $\frac{T_2}{T_1 - T_2}$, the refrigeration effect will be $= \frac{T_2}{T_1 - T_2} W$, and the

$$\text{Real efficiency} = \frac{Q_3}{\frac{T_2}{T_1 - T_2} W} = \frac{Q_3(T_1 - T_2)}{T_2 W}. \qquad (10)$$

439. Wet and Dry Compression.—The meaning of these terms has already been outlined in Art. 435. As far as relative efficiency of the two systems of operation is concerned, this has to be considered from a practical as well as theoretical standpoint. In theory, referring to Fig. 593, the work expended in wet compression is area *ABC′D′*, the useful effect is area *BC′GF*. In the dry compression system, starting at *C*, the work done is *ABCDD′A*, the useful effect is *BCG′F*. The work area is increased by *C′CDD′*, the useful effect by area *C′CG′G*. It will at once be seen that, for the two processes to have the same efficiency, it would have been necessary to increase the work area in dry compression only by the area *D′C′CR*. The extra area *DD′R* is therefore a measure of the loss in the dry compression system as compared with the wet. Even in theory this loss is, however, so small as to make no practical difference. It increases, of course, with the degree of superheat in the vapor at the suction valve, while for any compression starting between *C′* and *C* the loss is less.

In practice it is common in even the wet system to choose a point between *C* and *C′* so that the vapor is slightly superheated at the end of compression. Any other condition at this point would seriously cut down the refrigeration effect. As far as the practical merits of the two systems of operation are concerned, there has been much controversy that cannot be entered into here. Ewing comes to the conclusion that, everything considered, there is little difference in the efficiency either on theoretical or on practical grounds.

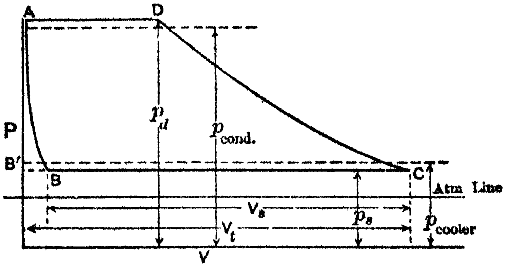

FIG. 596.—COMPRESSOR INDICATOR DIAGRAM, REFRIGERATING MACHINE.

440. The Compressor Indicator Diagram. Volumetric Efficiency.—The general shape of the compressor diagram is shown in Fig. 596. *AB* is the reëxpansion line, if there is vapor left in the clearance spaces at the end of the compressor stroke. From *B* to *C* the compressor draws vapor from the suction pipe. The suction pressure p_s is less than the cooler pressure p_{cooler} by the amount of wire drawing in the suction valve. Both on account of reëxpansion and the heating of the incoming vapor, the weight of charge per stroke is less than the piston displacement calls for at the pressure and temperature existing just ahead of the suction valve. The ratio of the actual charge weight per stroke to the weight computed on the basis of piston displacement for the pressure and temperature just ahead of the suction valve may be regarded as the *true volumetric efficiency* E_{vt} of the compressor cylinder. The determination of this efficiency requires a knowledge of the amount of vapor circulated.

The *apparent volumetric efficiency* E_{v_a} is determined directly from the card, for it is the ratio of the volume V_s to the volume V_t. In the case of very small machines, E_{v_a} may be as low as 0.6 and does not go above 0.8. For medium sized machines it may be from 0.8 to 0.9, and in large machines from 0.9 to 0.98.*

The nature of the compression line CD depends upon whether wet or dry compression is used. The pressure p_d at D is greater than the pressure in the condenser by the amount required to overcome the valve resistance.

There are three assumptions possible as to the state of the vapor at the point C:

(a) Vapor wet at C. The compression line may then be assumed to follow the adiabatic for wet vapors. It has been shown that this type of adiabatic for NH_3 can be closely represented theoretically by the equation

$$pV^{1.17} = \text{constant}.$$

Starting at point C, we can now draw in the theoretical compression line, and get some idea of the heat interchanges going on during compression by comparing this with the actual compression curve.

(b) Vapor superheated at C. In this case the theoretical comression adiabatic is closely enough represented by the equation

$$pV^{1.323} = \text{constant}.$$

(c) Vapor dry and saturated at C. In this case superheat immediately takes place theoretically, and the case is then the same as (*b*).

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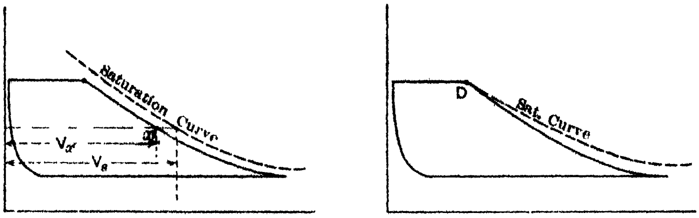
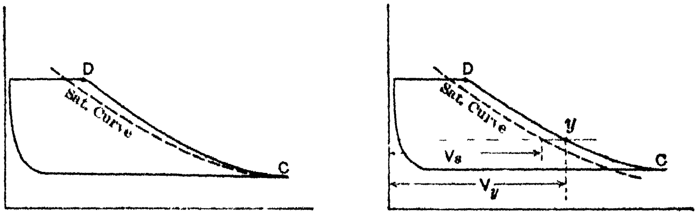
* Döderlein, Prüfung & Berechnung ausgeführter Ammoniak-Kompression Kältemaschinen.

From the amount of NH_3 in the cylinder per cycle (consisting of the sum of the NH_3 taken from the suction pipe per stroke plus the vapor, if any, caught in clearance), it is possible to construct a *saturation curve* by aid of the specific volumes, by the same method as for steam. The location that this curve may have relative to the compression line is indicated in Fig. 597 *a* to *d*. In case the compression line shows wet vapor, the quality of that vapor at any point may be computed as in the case of steam; that is, in Fig. 597*a*, quality at $x = \frac{V_x}{V_s}$.



Wet Compression-Vapor Wet throughout (*a*)

Wet Compression-Vapor dry at D (*b*)



Dry Compression-Vapor dry at C-Superheated at D (c)

Dry Compression Vapor Superheated at C and D

Fig. 597.—*a* to *d*.

If the vapor is superheated, the assumption that the vapor acts as a gas will have to be used in the present state of our knowledge; that is, in Fig. 597*d*, the absolute temperature of the vapor at point *y* may be found from the equation

$$\frac{T_y}{T_s} = \frac{V_y}{V_s},$$

in which T_s is the saturation temperature at the pressure chosen.

441. The Weight of NH_3 in Circulation. — The amount of NH_3 in circulation may be determined experimentally or computed approximately. It is not a necessary item for the computation of the amount of refrigeration or the energy expended, is therefore not required to determine the coefficient of performance, and is in consequence not often determined experimentally.

The direct determination of weight is made on the liquid (high-pressure) side of the machine. It has been done by using a piston meter, or other suitable type, and by interposing weighing tanks on scales between the condenser and the receiver. The former method was used by J. E. Denton in a test on a 75-ton machine, the test being reported in Trans. A. S. M. E., Vol. XII, 1891. The meter used was connected to the outlet of the condenser and was a ¾-inch Worthington piston meter. It is stated that the valves of this meter were specially made of wrought steel, and that a few extra bolts were used in the top and bottom joints. The meter was tested for tightness against 250 lbs. of air pressure under water. It gave no trouble during a month's use, except that the packing around the dial arbor had to be renewed.

When weighing tanks on scales are used, it is customary to use two connected in parallel in order to obtain a continuous reading. Each is fitted with valves on the inlet and outlet, so that each may be cut out of the system at will, and with a gauge glass so that the level of the liquid NH_3 may be readily determined. The tanks are connected by piping, on the one side to the condenser, on the other to the receiver, of such length that there is sufficient flexibility to give a satisfactory reading of difference on the scale. In the testing plant at Sibley College, the tanks are submerged in water to maintain a constant temperature, but this is probably not necessary. In using this weighing system, suppose that inlet valve to tank *A* and the outlet valve to tank *B* are closed, while the other valves are open. This means that tank *A* is discharging into the receiver while tank *B* is filling from the condenser. When tank *B* is filled, *A* will be down; then the connections are changed. The gross and tare weights recorded for each tank of course determine the total weight passing.

A single tank may be made to do service, except that a rate of flow rather than a continuous determination is made. In this case, with the outlet valve closed, the tank is allowed to fill up from the condenser. The inlet valve is then closed just long enough to take the gross weight, the outlet valve opened wide to quickly drain the tank contents into the receiver and to obtain the new tare weight with the outlet valve closed. The inlet valve is then opened and the operation repeated.

It is essential, in any direct system of weighing, to see that the amount of liquid present in any location in which liquid may gather is exactly the same in the end as at the beginning of the test. If this condition cannot be reached, proper allowances must be made.

Where a volume measurement of liquid NH_3 is made, the density must of course be known to obtain the weight. This is given by Denton* as follows:

Temp. ° F.	5	14	23	32	41	50	59	68	86
Density, Distilled Water = 1.00 = 62.43 lbs. per cu. ft.	.731	.6492	.6429	.6364	.6298	.6230	.616	.6089	.6018

Several more or less approximate methods of computation may be used to determine the amount of NH_3 in circulation.

The common way is to start with the apparent volumetric efficiency of the compressor (see Art. 440). If this is represented by E_{va} and the actual piston displacement is V cu. ft. per stroke, the effective displacement per stroke will be $E_{va}V$ cu. ft. If we let the volume of 1 lb. of the vapor at the end of the suction stroke be v cu. ft. per lb. and N the number of compressor cycles per hour, the weight of NH_3 in circulation per hour will evidently be

$$= \frac{E_{va}VN}{v} \text{ lbs.} \tag{11}$$

The uncertain factor in the computation is the quality of the vapor at the end of the suction stroke, if wet compression is used, or the temperature at the same point, if dry compression is employed. Both affect the value of v. It is certain that the vapor is drier (or more highly superheated) at the end of the suction stroke than it is ahead of the suction valve, owing to the heating effect of the cylinder walls.

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* Trans. A. S. M. E., Vol. XII, 1891.

Both DeVolson Wood* and J. E. Denton† have made computations on the latter point for *superheated* vapor.

The former, for a case in which the suction pressure was 28.9 lbs., the suction temperature 57.7°, and the outlet temperature 116.1°, assumes a temperature at the end of suction equal to 105°. Zeuner's equation for the specific volume of saturated or superheated NH_3 vapor is

$$v = \frac{0.667\,T - 4.9\,p^{.32}}{p}, \tag{12}$$

in which T = absolute temperature and p = absolute pressure in pounds per square inch. Using the values $T = 460 + 105 = 565°$ and $p = 28.9$ lbs., this equation gives $v = 12.4$ cu. ft. per lb. Using the values $T = 460 + 57.7 = 517.7°$ and $p = 28.9$ lbs., the value of $v = 11.4$ cu. ft. per lb. If the latter value is used, as is often done, the weight of NH_3 determined is too large by

$$\frac{12.4 - 11.4}{12.4} = 8.1 \text{ per cent.}$$

In the test reported by Denton, in which the NH_3 was actually metered, the computation on the basis of volumetric efficiency and specific volume at suction pressure and temperature showed a weight of NH_3 in circulation 21.4 per cent greater than the actual. Evidently there must have been considerable heating of the gas as it entered the cylinder, and Denton computed that the temperature at the end of the suction stroke must have been 80° instead of 14°, as shown just ahead of the suction valve. The suction pressure was in this case 42 lbs. abs.

It will be evident from the above that everything depends upon the proper assumption of temperature at the end of the suction stroke. If the vapor at the end of the suction stroke is still *wet*, the problem is even more difficult. In that case, Döderlein (see reference, p. 1022) makes the assumption that the vapor is just dry at the end of compression. Assuming that the process is isentropic, he computes the quality at suction pressure on the card from the relation

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* Trans A S. M. E., Vol. XI, 1890, p. 833.

† Trans. A. S. M. E., Vol. XII, 1891, p. 372

$$\phi_{C'} + x_{C'}\frac{r_{C'}}{T_{C'}} = \phi_{D'} + x_{D'}\frac{r_{D'}}{T_{D'}}, \tag{13}$$

in which $\phi_{C'}$ and $\phi_{D'}$ are the entropies of the liquid at C' and D' respectively, in Fig. 593.

$x_{C'}$ and $x_{D'}$ are the qualities of the vapor at C' and D', and $\frac{r}{T}$ represents the entropy of the dry vapor.

By assumption, $x_{D'} = 1.0$, and since the absolute pressures at C' and D' are known, the other quantities may be obtained from the NH_3 table, and $x_{C'}$ may be computed. The volume per pound of the vapor then follows from the relation

$$v = xu + \sigma = x(v' - \sigma) - \sigma = \text{approx. } xv', \tag{14}$$

in which x = quality of vapor.
u = increase in volume during vaporization at the pressure under consideration.
σ = specific volume of the liquid at the same pressure.
v' = specific volume of the dry and saturated vapor at the same pressure, obtained directly from vapor tables.

This value of v is then substituted in Eq. (11).

The computation above outlined requires, for the case of superheated vapor, an estimate of the temperature at C'', Fig. 593, while that for the wet vapor determines the quality at C', on the assumption that D' is on the saturation line. As an alternative method, it is possible to obtain an approximation to the weight of NH_3 in circulation by aid of the entropy diagram of the real cycle. The area of this diagram represents the work input per pound of NH_3. If, therefore, we divide the heat equivalent of the actual compressor horse-power per hour (= Comp. H.P. × 2545) by this heat input per pound of NH_3, the result will be the weight of NH_3 in circulation per hour. This method holds for the vapor under any condition of quality or superheat.

442. **Capacity and Rating of Refrigerating Machines.** — The amount of heat actually removed from the cold body by a refrig-

erating machine (in case brine circulation is used, equal to the weight of brine circulated per unit of time times range of temperature of cooling times specific heat) is known as the *cooling* or *refrigerating effect.* One measure of efficiency or economy would therefore be to state the cooling effect produced per compressor horse-power, or per steam horse-power, or per pound of coal used.

The heat of fusion of ice, from ice at 32° to water at 32°, equals 144* B.t.u. Hence to freeze 1 ton of water under these conditions requires $2000 \times 144 = 288{,}000$ B.t.u. This quantity is sometimes called the *unit* of *refrigeration.*

The cooling effect per day of 24 hours divided by the unit of refrigeration gives the so-called *ice-melting capacity* of the plant. The number of tons of ice-melting capacity is called the *tonnage* of the installation.

Sometimes the *ice-making capacity* is spoken of. This term has no definite meaning, for in any given plant the quantity of ice that can be made in any given time depends upon the temperature of the water to be frozen and upon the temperature of the ice leaving the cooler. Assume, for instance, that the water has a temperature at 60°, and that the ice is cooled to 20°. Taking the specific heat of ice at 0.5, this would mean heat removed per pound of water frozen $= (60 - 32) + 144 + \{0.5 \times (32 - 20)\} = 178$ B.t.u. The ice-making capacity is therefore only $\frac{144}{178} = 80.6$ per cent of the ice-melting capacity.

A secondary standard of economy also sometimes given in test reports is the amount of ice made per pound of fuel used. This is not definite for the reasons above mentioned and also because the quality of the coal differs from plant to plant. It cannot, therefore, be used as a general standard of comparison, and applies only to the plant in question.

443. Arrangements for Testing of Vapor-compression Refrigerating Machines.† – As in the case of a steam plant, there may be a number of objects in view in the testing of a refrigerating plant. The main objects, of course, generally are the determination of

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* Different authorities give from 142 to 144 B.t.u. for this figure.

† For a complete discussion of the testing of an absorption machine, see Trans. A. S. M. E., Vol. X, p. 792.

economy and *capacity;* but there are a number of secondary tests that may be made, such as for the efficiency of the cooler, the proper operation of the condenser, the action of the valves of the compressor, the efficiency of the compressor itself apart from the rest of the apparatus, etc. It will be assumed in what follows that a complete test for capacity and economy is to be made.

Each compressor refrigerating plant consists of several very distinct parts, on each of which certain measurements must be made. These are: the compressor with its motive power, the condenser, and the cooler or refrigerator.

Compressor and Prime Mover. — The usual source of motive power for the compressor is a steam engine, commonly of the low-speed type. The arrangements necessary for the determination of the power developed (I.H.P.) are fully taken up in Chap. XVIII, to which the student is referred.

The work done by the compressor (Compressor H.P.) is determined directly by means of indicator cards. The indicators must be of special make, that is, of steel, since brass, the metal ordinarily employed, is attacked by NH_3. The indicator connections should be as short as possible, to reduce clearance space, and each indicator should be fitted with a special quick-acting valve. The indicator pistons should fit fairly tight, to prevent the escape of the disagreeable gas, but of course they must not be so tight as to affect the accuracy of the instrument. The driving mechanism used for the indicator depends altogether upon the type of machine, the points made in Chap. XV also governing this case.

Other readings besides power determination required on the compressor are:

(*a*) Quantity of jacket water, if a cooling jacket is used.

(*b*) Temperatures of jacket water in and out.

(*c*) Temperatures of NH_3 vapor just ahead of the suction valve and just beyond discharge valve.

(*d*) Pressure of NH_3 vapor in the same places in which temperatures are measured.

(*e*) Speed of compressor.

(*f*) Quantity of oil fed to cylinder, if that system of eliminating clearance is used.

(*g*) If possible, quantity of liquid NH_3 injected, if the wet system is used.

The Condenser.—The readings on the condenser are the following:

(*a*) Quantity of condensing water, best obtained by meter.

(*b*) Temperature of condensing water, in and out.

(*c*) Temperature of NH_3 vapor in and NH_3 liquid out.

Between the cooler and the expansion valve is interposed the means for determining the quantity of NH_3 in circulation (see Art 441).

The Cooler.— Readings required, assuming that brine circulation is used:

(*a*) Quantity of brine in circulation, best obtained by meter.

(*b*) Temperature of brine in and out of cooler.

(*c*) Specific gravity of brine.

(*d*) Temperature of NH_3 entering or leaving cooler.

When a plant is to be tested for capacity and the conditions are such that the heat taken up by the brine in doing its normal work is not sufficient to heat back to the upper temperature, means must be provided to supply the extra heat. This can usually be done by immersing a part of the brine coil in water, which may be heated by steam. No account need be taken of the amount of heat so supplied.

The log blanks, pages 1031 and 1032, for recording observed data were constructed primarily for the 15-ton York machine used at Sibley College, but show a general and concise arrangement of the readings necessary.

The greatest difficulty to contend with in the making of an accurate test of a refrigerating plant consists in the fact that it is very difficult to be absolutely certain that all parts of the plant, particularly the cooler, contain the same amount of heat, estimated above some datum. at the end as at the beginning of the test. To reduce the possible error in this respect, two things are necessary: first, keep all temperatures the same as nearly as possible; and second, prolong the test over a considerable period of time. The latter should never be less than 12 hours, and may preferably be several times that length.

Test of Refrigerating Plant built by. . .. . *Date* . . . .

Steam Engine
Horizontal, Double-acting

Piston Rod.
Diameter . . . . .
Area. . . . . . .

Cylinder.
Stroke .
Diameter. . . .
Area { H
Area { Cr . . . .

Ammonia Compressor.
Vertical, Single-acting.

Cylinder A
Stroke .
Diameter. . . . . .
Area. . . . . . .
Volume. . . .

Cylinder B.
Stroke. . . . .
Diameter . . . . .
Area. . . . .
Volume . . . . . .

Brine Pump.
Stroke. . . . .
Diameter. . . . .
Area . . . .
No. of Cylinders . . . .
Volume . . . .

Time.	Barometer.	Temperature.		Stroke Counter	Revolutions per Min.	Steam Pressure.	Ammonia Pressure		Temperature				Pounds of Jacket Water, per hour	Temperature.				Pounds of Jacket Water, per hour	Stroke Counter.	Revolutions per Min.	Pressure.		
									Ammonia.		Jacket Water			Ammonia		Jacket Water.							
		Room.	External Air.				Suction.	Discharge	Suction	Discharge	Entering.	Leaving.		Suction.	Discharge	Entering	Leaving				Suction.	Discharge	
. . . .																							. . . . . .
. . . .																							. . . .
. . . . . .																							. . . . . .
. . . . .																							. . . . . .
Average																							. . . .

Test of Refrigerating Machine built by . . . . *Date* . . . .

Brine Cooler. Specific Gravity of Brine.. ..										Ammonia Condenser.												
Time	Temperatures								Weight or Volume of Brine.	Time.	Temperatures								Weight or Volume of Condensing Water.	Weight or Volume of Liquid NH_3	At NH_3 Weighing System.	
	Coil No I.				Coil No. II						Coil No. I.				Coil No. II							
	NH_3.		Brine.		NH_3		Brine				NH_3.		Cond. Water.		NH_3		Cond Water.					
	In.	Out.	In.	Out	In	Out	In.	Out.			In.	Out	In	Out	In	Out.	In.	Out.			Press	Temp.

The frequency of taking readings depends upon conditions. Where these are fairly constant, half-hour readings are probably quite sufficient. Where this is not the case, the number of readings must be increased, but the matter must be left to the judgment of the testing engineer.

444. Computation of Results. — The forms on pages 1034 and 1035 contain the items of computation for a complete test. The first of these shows average pressures, temperatures, weights, etc., computed from observed data; the second contains the final results.

In view of the explanations contained in previous articles in this chapter, but few of the items on the two forms need any further comment or recapitulation.

Item (63). Compressor horse-power is computed from the compressor-indicator cards in exactly the same way as I.H.P. for a steam engine.

Items (65) to (71). See Art. 440.

Item (88). Mechanical efficiency of set $= \frac{\text{Item 63}}{\text{Item 60}}$.

Item (89). Actual coefficient of performance $= \frac{\text{Item 81}}{\text{Item 64}}$. See Art. 438.

Item (90.) Coefficient of performance of set $= \frac{\text{Item 81}}{\text{Item 62}}$.

Item (91). Theoretical coefficient of performance. See Art. 436.

Item (92). Efficiency of refrigeration process = Actual coefficient of performance ÷ Ideal coefficient of performance. See Art. 438.

Item (93). Tonnage. See Art. 442.

445. The Heat Balance for the Refrigeration Process. — This balance is established by following the refrigerating agent through the complete cycle, balancing the sum of all the amounts of heat received by the agent against the sum of the amounts of heat given up by it.

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Test of Refrigerating Plant built by . . . . *Date* . . . .

AVERAGE DATA, OBSERVED OR COMPUTED.

1. Duration of test, hours. . . .

Dimensions, Volumes, Etc.

2. Diam steam cyl., in....
3. Stroke, steam cyl., in..
4. Diam. piston rod, steam cyl., in.
5. Area steam cyl., head, sq. in.
6. Area steam cyl., crank, sq. in...
7. Diam. compressor cyl. A, in
8. Diam. compressor cyl. B, in
9. Stroke, compressor cyl. A, in..
10. Stroke, compressor cyl. B, in.. .
11. Piston displacement, compressor cyl. A, cu. ft.
12. Piston displacement, compressor cyl. B, cu. ft.
13. Diam. brine pump, in
14. Stroke, brine pump, in
15. No. of cyls. brine pump
16. Piston displacement per cyl., brine pump, cu. ft . .

Revolutions per Minute.

17. Steam engine . .
18. Compressor. . .
19. Brine pump ..

Pressures.

20. Steam pressure, lbs....
21. Back press. on steam calorimeter, ″ Hg. . .. .
22. Condenser press., ″ Hg. . .
23. Barometer, ″ Hg
24. Suction press., compressor, lbs .
25. Discharge press , compressor, lbs.
26. Suction press , brine pump, lbs.
27. Discharge press., brine pump, lbs.
28. Pressure of liquid NH_3 at weighing system, lbs. . .

Temperatures, Deg. Fahr.

29. External air.... . . . .
30. Room.
31. Condensing water, steam engine, inlet . . . . . .
32. Condensing water, steam engine, outlet.. . .. .
33. Condensed steam. . .
34. Temp. in steam calorimeter.. ..
35. Suction temp. NH_3. .
36. Discharge temp. NH_3....
37. Jacket water, compressor, inlet..
38. Jacket water, compressor, outlet.
39. Condensing water, NH_3 condenser, inlet
40. Condensing water, NH_3 condenser, outlet
41. Temp of NH_3, inlet to condenser
42. Temp. of NH_3, outlet from condenser
43. Temp. of brine, inlet to cooler.
44. Temp. of brine, outlet from cooler.
45. Temp of NH_3, inlet to cooler
46. Temp of NH_3 outlet from cooler.
47. Temp. of NH_3, in front of expansion valve .
48. Temp. of liquid NH_3 at weighing system...

Weights of Water and NH_3, per Hour

49. Weight of steam... ..
50. Weight of NH_3 in circulation if measured directly..... ...
51. Weight of jacket water, compr. .
52. Weight of water, NH_3 condenser.
53. Weight of condensing water, steam engine .. . .
54. Weight of brine in circulation

General Data

55. Quality of steam . . ..
56. Specific heat of brine. . .
57. Specific gravity of brine. . .
58. Square feet of NH_3 condenser surface . .. .
59. Square feet of cooler surface . . .

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Test of Refrigerating Plant built by . . . . *Date* . . . . . .

FINAL RESULTS.

Steam Engine.

60. I.H.P. . . . .
61. Water rate per I H.P. hour . . .
62. B.t.u. supplied per hour . . . .

Compressor.

63. I H.P. . . . .
64. B.t u. hour equivalent of I.H.P.
65. Apparent volumetric efficiency, E_{va} . . . . .
66. Piston displacement, cu. ft. .
67. Effective piston displacement, cu. ft. . . .
68. Specific volume of NH_3 at suction pressure and temperature. . .
69. Weight of NH_3 per stroke, lbs
70. Weight of NH_3 per hour, lbs., computed. . . . . .
71. Ratio of weight of NH_3 computed to actual weight
72. Range of temp., jacket water
73. B.t.u. per hour in jacket water .

Condenser

74. Weight of water per hour. . .
75. Range of temperature, water. .
76. B.t.u. per hour in condensing water . . . .
77. Total heat removed from each pound of NH_3. . . . .
78. Heat removed from NH_3 per hr. .

Cooler.

79. Weight of brine per hour. . . .
80. Temperature range of brine. . . .
81. B.t.u. per hour from brine. . . .
82. Heat given to each pound of NH_3
83. B.t.u. per hour to NH_3. . . . .
84. Efficiency of cooler . . . .

Brine Pump.

85. Work done by pump. . . . .
86. Power input to pump . . . .
87. Mechanical efficiency . . . .

General Results.

88. Mechanical efficiency of set
89. Actual coefficient of performance
90. Coefficient of performance of set
91. Theoretical coefficient of performance
92. Efficiency of refrigeration process
93. Tonnage . . .
94. Tonnage per pound of NH_3 . .
95. Tonnage per steam I.H.P. . . . . . .
96. Tonnage per lb. of steam used.
97. Gallons of condensing water per ton of refrigeration . . . . .

A. *Heat Received by Agent per Hour.*	B.t.u.	Per cent of Total.
1. Heat taken from brine (= Item 81).	. . . . .	. . . . .
2. Heat due to radiation in cooler (= Item 83 − Item 81).	. . . . .	. . . . .
3. Heat due to radiation in piping, cooler to compressor.	. . . . .	. . . . .
4. Heat equivalent of compressor I.H.P.	. . . . .	. . . . .
A =	. . . . .	. . . . .

B. Heat Given up by Agent per Hour.	B.t.u.	Per cent of Total.
1. Heat removed by compressor jacket (= Item 73).....................	.....	.....
2. Heat radiated from piping to condenser........................	.....	.....
3. Heat removed in condenser (= Item 77 × lbs. of NH_3 in circulation per hour)......................	.....	.....
4. Heat radiated from piping, condenser to expansion valve...........	.....	.....
$B =$	.....	.....

CHAPTER XXV.

HYDRAULIC MACHINERY.*

446. Classification. — The broadest classification divides all hydraulic machinery into two classes: *hydraulic motors* and *pumps*. Machines belonging to the former class are prime movers and take energy imparted to them by water, converting a part of it into other forms of mechanical energy. Pumps, on the other hand, impart energy to water, being operated by some type of prime mover.

Hydraulic motors are again divided into the following classes:

(*A*) Water-bucket engines, in which water is allowed to flow into suspended buckets,. which in their descent lift weights and overcome resistances. This type is practically obsolete and will not be considered further.

(*B*) Hydraulic rams and jet pumps, in which the energy of one mass of water is utilized to impart energy to a second mass. These machines may also be looked upon as pumps.

(*C*) Water-pressure engines, using the direct pressure of water against moving machine members. The latter may be either reciprocating or rotary.

(*D*) Water wheels, with horizontal shaft, in which the water may act either by weight or by impulse, or by a combination of both. This class includes both the old-fashioned water wheels and the impulse wheel.

(*E*) Turbines, rotating either about a vertical or a horizontal shaft, in which the water acts by pressure and by impulse.

The different classes of pumps correspond broadly to the different classes of water motors, with the mechanical principles of operation reversed. Thus the reciprocating pump corresponds to the water-pressure engine; chain-and-bucket pumps to water wheels, in which the water acts principally by weight; centrifugal pumps are closely analogous to turbines. A broad general classification is into

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* For a general study of this wide and very important field, the student is referred to I. P. Church, Hydraulic Motors; Bovey, Hydraulics; Merriman, Treatise on Hydraulics; F. C. Lea, Hydraulics.

(*A*) Reciprocating pumps and
(*B*) Rotary pumps.

The former class includes steam pumps, direct water-pressure pumps, etc. Rotary pumps may again be subdivided into two distinct kinds: (*a*) centrifugal pumps and (*b*) rotary pumps in the narrower sense of the word. The working parts of these latter pumps are usually gears or cams meshed together.

Pumps are also sometimes classified according to the power driving them. Thus we distinguish steam pumps, motor-driven pumps, power pumps if driven from shafting, etc.

Three other types of pumps will have to be mentioned. The first of these is the pulsometer, which uses the direct action of steam to pump water; the second is the Humphrey gas pump, in which the explosive force of a combustible gas mixture is directly used; and the third is the method of pumping by compressed air.

Hydraulic Motors.

447. The Hydraulic Ram. — This is a machine which uses the momentum of a stream of water falling through a small height to raise a part of the water to a greater height.

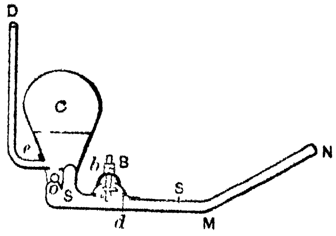

FIG. 598. — HYDRAULIC RAM.

Fig. 598 shows a conventional sketch of a simple type of ram. It consists essentially of an air chamber C connected to the discharge pipe eD, a check or delivery valve O, a waste or clack valve Bd, and a supply chamber SS. MN is the supply or drive pipe.

The action may be explained as follows: Assume that the waste valve is open and the delivery valve closed. This is always the case when the water is at rest, for gravity alone will open the first valve and close the second. If now the water starts to flow, it will escape past the waste valve, acquiring greater and greater velocity, until the pressure difference between the under and upper sides of the valve becomes sufficiently great to close the valve abruptly against the action of gravity. The sudden closing of the valve arrests the motion of the water and causes a rapid rise of pressure in the supply chamber *SS* to such an extent as to lift the delivery valve *O* against the resistance acting on the other side. A portion of the water thus passes into *C*, compressing the air in it. The recoil of the water in the chamber *SS* next causes the pressure to fall again, so that the delivery valve closes and the waste valve opens, after which the operation is repeated. While the water in the drive pipe is accumulating velocity for the next operation, the air in *C* expands, forcing the water up to delivery pipe *eD*. The flow up *eD* is fairly constant if a large enough air chamber *C* is provided. To keep the air chamber properly filled with air, a small snifting valve, not shown in Fig. 598, opening inward below the delivery valve, is provided, through which, when the pressure on the recoil falls below atmospheric, a small quantity of air is drawn in. This air is delivered to the chamber *C* along with a certain quantity of water on the next forcing operation. The waste-valve stem is furnished with an adjustment at *b*, so that the length of stroke may be changed.

The drive pipe should be straight and as free from friction as possible. It must be of a certain minimum length for satisfactory operation; a length not less than five times the supply head is recommended.

448. The Efficiency of the Hydraulic Ram. — In Fig. 599 the ram *R* receives its supply water from a reservoir *A* at a head h_s above the level of the waste valve, and delivers a part of it into a reservoir *B* at a height h_d above the level of the valve. Let W_s = weight of water passing down the supply pipe in a given time, W_w = the weight of water wasted at the waste valve, and W_d = weight of water delivered. There are two methods of computing the efficiency of a ram. The first considers that the useful work consists in lifting W_d pounds of water a distance $h_d - h_s$ above the level of the supply. The useful work, therefore, is $W_d\ (h_d - h_s)$ foot-pounds. It also considers that the total power expended to do that work is that due to the waste water W_w falling through the supply head h_s, that is, $W_w h_s$. From this we have

$$\text{Efficiency} = \frac{W_d\,(h_d - h_s)}{W_w h_s}. \tag{1}$$

The other method states that the total energy expended is $W_s h_s$, and that the useful effect is $W_d h_d$, so that

$$\text{Efficiency} = \frac{W_d h_d}{W_s h_s}. \tag{2}$$

The first equation for efficiency is known as the Rankine formula and is the one mostly used. The last form is due to d'Aubisson.

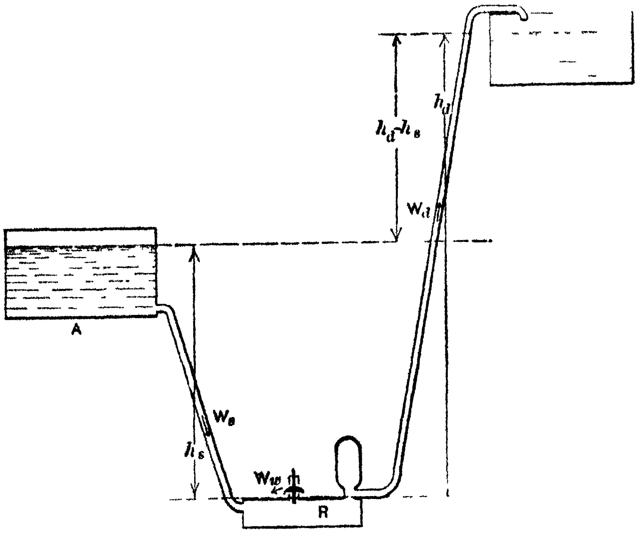

FIG 599 — HYDRAULIC RAM AND CONNECTIONS.

449. Testing of Hydraulic Rams. — A test of an hydraulic ram requires the following observations: (*a*) supply head, (*b*) discharge head, (*c*) waste water, (*d*) water pumped, (*e*) length of stroke (or spring setting) of waste valve and strokes per minute of this valve, (*f*) dimensions of the ram and details as to length and course of supply and discharge pipes.

Where the supply and discharge pressures are small, the best instrument with which to measure them is the ordinary manometer.

In the ordinary case, however, a gauge must be used for the discharge head. Convert the head in pounds to feet, and add the distance from the center of the gauge to the level of the waste valve, if the gauge is located above the ram, as it usually is. Except for very large rams, both waste water and water pumped may generally be directly weighed.

The independent variables that may be changed in the operation of a ram are (*a*) the supply head, (*b*) the discharge head, and (*c*) the number of strokes in unit time of the waste valve. The latter variable is adjusted by changing the length of the stroke of the waste valve, or the tension of the waste-valve spring.

The following series of tests may be made:

(*A*) With setting of waste valve constant, take a series of supply heads, and for each supply head change the discharge head by a series of steps from a minimum to the maximum. The discharge head is most easily adjusted by putting a throttle valve above the gauge in the discharge pipe. The action of this valve is equivalent to adding or subtracting from the lift, as the case may be.

(*B*) For any setting of the supply and discharge heads, make a series of runs by changing the setting of the waste valve.

Each run should last about one-half hour. The form on page 1042 shows the data to be recorded, together with the quantities to be computed.

In the report, plot curves for each setting of waste valve between discharge heads h_d as ordinates and the following abscissæ: (*a*) capacity in gallons per twenty-four hours; (*b*) efficiency (Rankine); (*c*) waste water per twenty-four hours (gallons).

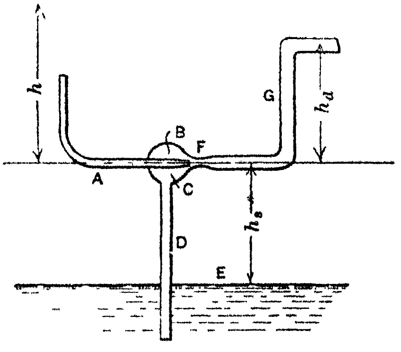

Fig. 600 — Jet Pump.

450. The Jet Pump. — The principle of operation of the jet pump may be explained from Fig. 600. Water under pressure is supplied through the pipe *A*, which ends in the nozzle *B* in the suction chamber *C*. The suction produced in *C* raises the water through *D* from the source of supply *E*, and the combined streams of high-

DEPARTMENT OF EXPERIMENTAL ENGINEERING—SIBLEY COLLEGE.

TEST OF HYDRAULIC RAM.

Made by . . . . *Date* 191

. . . .

Dimensions:

Air Chamber . . . . . . *Volume* . . .

Water-valve Diam. . . . *Area.* . . . . . .

Supply Pipe. . . . *in diam.; length* .

Number of Bends, etc., in Supply Pipe . .

Discharge Pipe . . . *in diam.; length*

Observers:

Number of Run.	1	2	3	4	5	6	7	8
Length stroke, inches. .								
Strokes per minute.								
Supply head.								
Supply head, feet of water, corrected, h_s								
Discharge pressure								
Discharge head, feet of water, corrected, h_d								
Ratio discharge to supply head								
Time, end of run. . . .								
Time, beginning of run								
Duration of run, minutes. . . .								
Water pumped, scale reading, end. . lbs.								
scale reading, beginning lbs.								
weight, W_d lbs.								
Water wasted, weight, W_w lbs.								
Total water supplied, $W_s = W_d + W_w$ lbs.								
Available energy (Rankine), $W_w h_s$ ft.-lbs.								
Work done (Rankine), $W_d(h_d - h_s)$. ft.-lbs.								
Work done per stroke. . . ft.-lbs.								
Capacity, pounds per minute.								
Capacity, gallons per 24 hours. .								
Efficiency, per cent (Rankine), $\frac{W_d(h_d - h_s)}{W_w h_s}$								
Efficiency, per cent (d'Aubisson), $\frac{W_d h_d}{W_s h_s}$								

pressure and suction water pass through the combining tube F and the discharge pipe G either to a reservoir or to waste. Such pumps are used for small lifts for many purposes, because they contain no working parts, although their efficiency is quite low, not exceeding 20 per cent.

A modification of the jet pump is the suction pump so extensively used in the chemical laboratory. Here the operating fluid is water, but the fluid pumped is air. Another modification is the steam injector (see Chap. XX), in which the operating fluid is steam and the fluid pumped is water.

451. Efficiency of the Jet Pump. — Let h be the effective head in feet of the supply water, h_s the actual suction lift to the center of the nozzle, and h_d the actual discharge lift; also let W = weight of supply water in a given time and W_s the weight of the suction water. Then energy available is $W(h - h_d)$ foot-pounds, and useful work is $W_s(h_s + h_d)$ foot-pounds, from which

$$\text{Efficiency} = \frac{W_s(h_s + h_d)}{W(h - h_d)}. \qquad (3)$$

452. Water-pressure Engines.—Appliances of this type (hydraulic cylinders, hydraulic jacks and hoists, etc.) are largely used where slow motion is desirable and high water pressure is available. The latter is generally obtained by means of steam pumps pumping into an accumulator.

The action of a water-pressure engine, of the type that operates continuously, as distinguished from that which operates intermittently, like the hydraulic jack or hoist, is very much the same as that of the steam engine, except, of course, that the fluid is used nonexpansively. The admission of the water to and the exit from the power cylinder may be controlled by slide and piston valves exactly as in the steam engine. An interesting type that has lately found application for small powers is shown in sketch in Fig. 601.

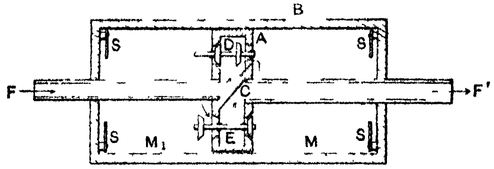

FIG. 601.—WATER PRESSURE ENGINE

A is a reciprocating hollow piston in a cylinder B. The piston is divided into two compartments by means of the partition C. The two admission valves rigidly connected on the same stem are shown at D, and the two exhaust valves, fastened in the same way, are shown at E. The stems are of such length that when one valve is against its seat the other valve on the same stem is off its seat.

The piston and tail rods are hollow. If water is admitted at F, it flows into the upper compartment of the piston, passes the right-hand inlet valve and reaches the end M of the cylinder, tending to force the piston to the left. At the same time the left-hand exhaust valve is open and allows the water on that side to flow into the lower compartment and so out through the rod at F'. When the piston approaches the end of the stroke, springs SS on the cylinder heads come into contact with the valve stems, are compressed until their resistance overcomes the water pressure holding the valves against the seats, when these valves are thrown over and the back stroke commences.

453. Efficiency of Water-pressure Engines.* — In any water-pressure engine A (Fig. 602), consider the stroke of the piston P from left to right. Let h_s be the equivalent total static supply head in feet above the level MN, as shown by a gauge G. It is assumed that the motion is slow and uniform against a resistance R. The unit pressure p_m (pounds per square foot) is somewhat less than the pressure due to h_s on account of entrance losses. Suppose it is equal to a head $h_{s'}$. The unit pressure $p_{m'}$, on the other side is somewhat greater than atmosphere on account of friction losses in the exit. Suppose it is equivalent to a head h_d. (It is here assumed that the exit water discharges into the air at the level MN.) If p_a = atmospheric pressure, then

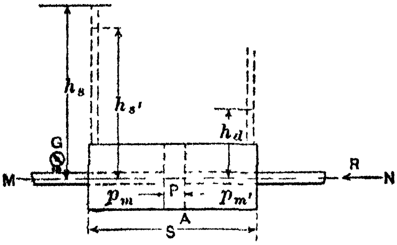

FIG. 602.

$$p_m = p_a + h_{s'}\delta, \quad \text{and} \quad p_{m'} = p_a + h_d\delta,$$

where δ = weight of one cubic foot of water. If the area of the piston is F square feet, we have for steady motion

$$R = F(p_m - p_{m'}) = F\delta(h_{s'} - h_d) \text{ lbs.} \tag{4}$$

Therefore the work done per stroke, if S is the stroke in feet, will be

$$RS = F\delta S(h_{s'} - h_d) \text{ ft.-lbs.}, \tag{5}$$

---

* See I. P. Church, Hydraulic Motors.

and if n = number of strokes per minute, the horse power developed under the conditions assumed will be

$$\text{H.P.} = \frac{F\delta S\,(h_{s'} - h_d)\,n}{33{,}000}. \tag{6}$$

$FSn\delta$ = weight W_s of water used per minute. Since h_s is the supply head, the available energy is $W_s h_s$ foot-pounds per minute. The hydraulic efficiency of the engine consequently is

$$E_h = \frac{F\delta Sn\,(h_{s'} - h_d)}{W_s h_s} = \frac{h_{s'} - h_d}{h_s}. \tag{7}$$

In practice it would be difficult to determine $h_{s'}$ and h_d on account of pressure fluctuations, but indicator cards may be taken from both ends of the cylinder. If the water horse power thus shown is I.H.P._w, the cylinder efficiency may be stated to be

$$E_c = \frac{\text{I.H.P.}_w \times 33{,}000}{W_s h_s}, \tag{8}$$

and if the piston speed is v feet per minute ($= nS$), so that the actual work done per minute is Rv foot-pounds, the total efficiency will be

$$E_t = \frac{Rv}{W_s h_s}. \tag{9}$$

454. **Vertical Water Wheels.** — This includes water wheels of the old type and impulse wheels. The former are again divided into *overshot* wheels, *breast* wheels, and *undershot* wheels.

455. **The Overshot Water Wheel.** — A general view of one of these wheels, explaining the action, is shown in Fig. 603. The water flows from a head race A into buckets around the circumference of the wheel and is discharged near the bottom. The head of water at the top must be such as to give the falling water greater velocity than that of the periphery of the wheel. In practice the velocity of the water is from 9 to 12 feet per second, that of the wheel from 5 to 10 feet per second. This type of wheel is not adapted to run in back water, and the greatest efficiency for any given head will be found when the wheel just clears the water in the tail race.

These wheels have been used with falls from 8 to 70 feet, and deliveries of from 3 to 25 cubic feet per second. When at their best they will show efficiencies of from 70 to 85 per cent. They are now, however, practically obsolete For that reason no derivation of work and efficiency formulas will be given, stating merely the final results.*

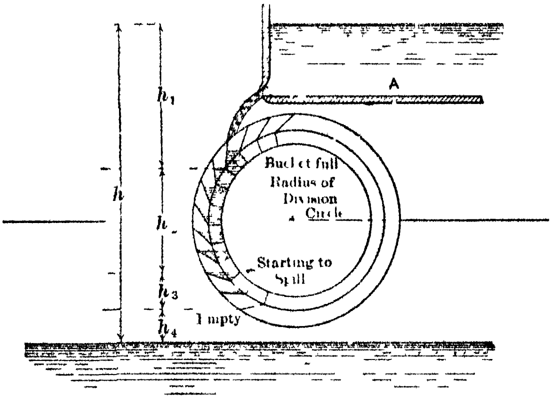

FIG 603 —OVERSHOT WATER WHEEL

The water acts on the wheel both by gravity and by impact. The power derived from the former source is much greater than that from the latter Church gives the following relation.

Power due to weight of water

$$= L_g = Q\delta\,[h_2 + \beta h_3]\ \text{ft.-lbs. per sec.} \qquad (10)$$

Power due to impact

$$= L_i = \frac{Q\delta}{g}[c_1 \cos \alpha - v_1]\, v_1\ \text{ft.-lbs. per sec.} \qquad (11)$$

Total theoretical power $= (L_g + L_i)$ ft.-lbs per sec. (12)

Total energy supplied $= Q\delta h$ ft.-lbs per sec. (13)

$$\text{Theoretical efficiency} = E = \frac{L_g + L_i}{Q\delta h}. \qquad (14)$$

For the significance of most of the symbols used, see the detail Figs. 603 and 604.

Q = cubic feet of water supplied per second, δ = weight per cubic foot of water. During the period that the bucket is passing through the vertical head h_3, the average weight of water contained in it is only some fraction β of the weight when full. β may be = .5, v_1 is the tangential velocity at the division circle, c_1 the absolute velocity of the jet, α the angle included between them.

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* For details see any of the books mentioned in footnote, p. 1037.

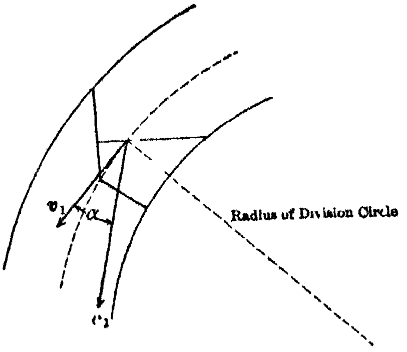

FIG 604

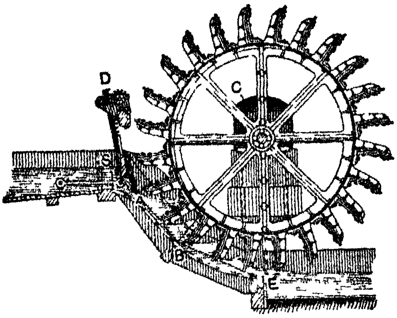

FIG 605 — BREAST WATER WHEEL

456. **The Breast Wheel.** — Fig. 605 gives a general view of this type of wheel. The water may be supplied by an overfall weir or through a sluice weir having a gate, as in Fig. 605. The apron of the flume *ABE* should fit the circumference of the buckets closely ($\frac{1}{2}''$ to $1''$) to prevent the loss of water. The buckets may be radial or turned backward to prevent losses on leaving the back water. Breast wheels have been used in falls from 5 to 15 feet with water supplies of from 5 to 80 cubic feet per second.

The power of the breast wheel is due to impact and gravity, as in the case of the overshot wheel, and essentially the same formulas apply. The total theoretical power may be expressed by

$$L = Q\delta\left[\frac{(c_1 \cos\alpha - v_1)}{g} + \beta h_2\right] \text{ft.-lbs. per sec.,} \qquad (15)$$

in which the symbols are the same as for the previous article. If h again denote the total head from the surface of the water in the head race to that in the tail race, and h_1 is the vertical distance from the surface of head water to the point of impact on the float, $h_2 = h - h_1$. The value of β depends upon how closely the apron fits the wheel; it may be as high as .9 when the fit is very close.

The theoretical efficiency is, as before,

$$E = \frac{L}{Q\delta h}. \tag{16}$$

The actual efficiency should range from 65 to 75 per cent.

457. The Undershot Wheel. — See Fig. 606 for a general view of an undershot wheel. These wheels take and reject the water at about the same level. The water enters the guide at a certain velocity due to its head to the surface of the head water and leaves it with a velocity about that of the tip of the floats. The power developed is almost wholly due to impact.

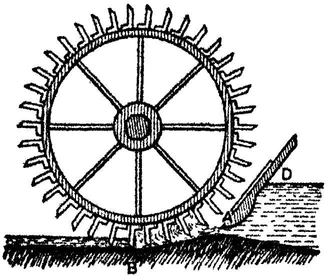

FIG. 606. — UNDERSHOT WATER WHEEL.

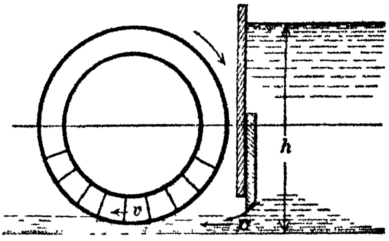

FIG. 607.

The equations for this wheel are as follows:* Let Q' = cubic feet of water per second actually suffering impact, Q = total number of cubic feet per second delivered through the gate, h the supply head in feet, and v the velocity of the center of the float (see Fig. 607). Then since impact is the only agency at work, equation (11), p. 1046, applies, and remembering that $\alpha = 0$ (since c and v are in the same direction), we know that the total theoretical power is

$$L = \frac{Q'\delta}{g}(c - v)\,v \text{ ft.-lbs. per sec.} \tag{17}$$

In this case, c is theoretically equal to $\sqrt{2gh}$. It can be shown that L is a maximum when $v = \frac{1}{2}c$, but that even then it equals only half the kinetic energy available in the water originally. Therefore, even if $Q' = Q$, the theoretical efficiency

$$E = \frac{L}{Q\delta h} \tag{18}$$

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* See Church, Hydraulic Motors.

could not exceed 50 per cent. In actual practice, the actual efficiency is from 25 to 33 per cent.

The *Poncelet* wheel is a modification of the undershot wheel in which the floats are concave toward the entering water. This increases the power to some extent, so that actual efficiencies of 68 per cent have been reached.

458. The Impulse Wheel or Tangential Water Wheel. — In this class of hydraulic motors one or more jets of water are made to impinge upon buckets fastened to the circumference of a wheel as they are successively brought into position by rotation. This class is very efficient for high heads of supply and small quantities of water.

The best-known impulse wheels are the Pelton, the Doble, and the Leffel (Cascade).* In all these wheels the buckets are characterized by a dividing ridge or midriff, as shown in Fig. 608,† which shows the running parts of a small Doble wheel. The tip *A* projecting from the nozzle *B* is the needle-regulating valve. A small laboratory motor complete (one side of the case glass) is shown in Fig. 609.

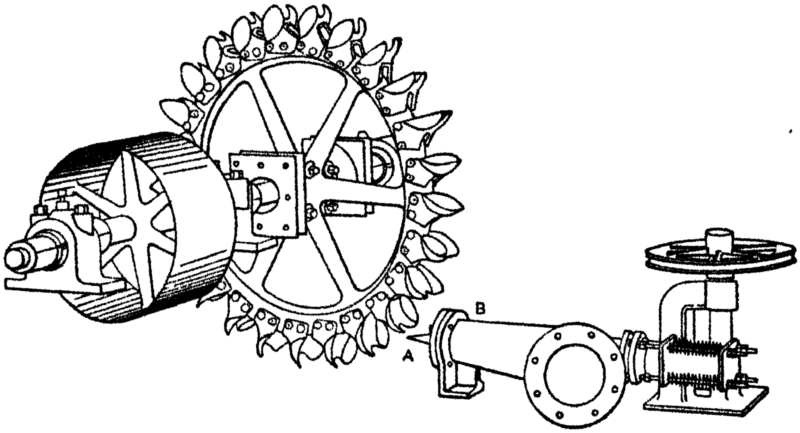

FIG. 608. — RUNNING PARTS OF AN IMPULSE WHEEL.

Since most of the wheels have curved vanes or buckets, this will

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* There is one other different type of impulse wheel (more strictly a turbine), the Girard, for the theory of which the student is referred to books on Hydraulics.

† Both this and the following figure are from the catalogue of the company.

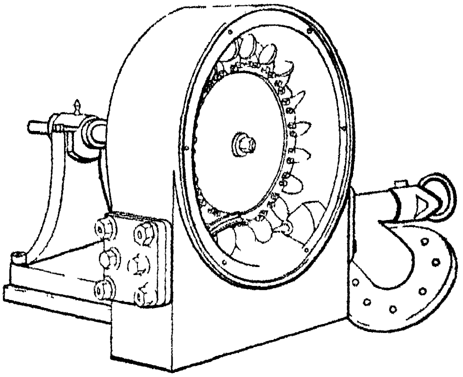

FIG. 609.—DOBLE IMPULSE WATER WHEEL

be the only case considered. Fig. 610 shows a cross section of one of these buckets, moving to the right with the velocity v, while the impinging stream has the velocity v_1, greater than v. The relative velocity of the particles of water leaving the bucket at A or B,

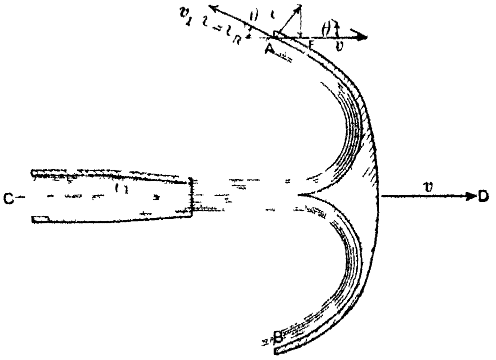

FIG. 610

neglecting hydraulic friction losses, is $v_1 - v = v_R$ in a direction of the tangent to the bucket element at A or B. The absolute velocity v_2 of the particle leaving the bucket at A is evidently the resultant of v and v_R, in a direction indicated by v_2. The component of the absolute velocity in the direction of the axis CD is AF, which is equal to $v - v_R \cos\theta = v - (v_1 - v) \cos\theta$. The change in the absolute velocity of a small mass ΔM of water in the direction of the axis CD consequently is $v_1 - [v - (v_1 - v) \cos\theta]$ or $(v_1 - v)(1 + \cos\theta)$.

Now since the velocity in the direction of the axis has been decreased, there must have been a *negative* acceleration in the time Δt, in the direction of v_1, equal to $p = \frac{(v_1 - v)(1 + \cos\theta)}{\Delta t}$. Since acceleration is also equal to $\frac{\text{Force}}{\text{Mass}} = \frac{P}{M}$, we may then write

$$p = \frac{P}{\Delta M} = \frac{(v_1 - v)(1 + \cos\theta)}{\Delta t}; \tag{19}$$

or the pressure exerted on the bucket by the small mass of water ΔM striking the bucket in the time Δt is

$$P = \Delta M \frac{(v_1 - v)(1 + \cos\theta)}{\Delta t} \text{lbs.} \tag{20}$$

The mass $\Delta M = \frac{Q'\delta}{g}\Delta t$, in which Q' is the volume in cubic feet of water striking the bucket in the time Δt, and δ is the density of the water in pounds per square foot. Hence

$$P = \frac{Q'\delta}{g}(v_1 - v)(1 + \cos\theta), \tag{21}$$

and the theoretical power developed by this force will be

$$L = Pv = \frac{Q'\delta}{g}(v_1 - v)(1 + \cos\theta)\, v \text{ ft.-lbs. per sec.} \tag{22}$$

Note that here Q' is not the amount of water issuing from the nozzle, but only that striking the bucket per second. That is, if F is the cross section of the nozzle, $Q' = F(v_1 - v)$, while the water issuing from the nozzle would be $Q = Fv_1$. But if a series of buckets be employed, as is of course the case in an actual wheel, the proportion of Q' used will increase, and finally we may put $Q' = Q$.

For any given value of v, the value of L is a maximum when $\theta = 0$, in which case $\cos\theta = 1.0$, and

$$L = \frac{2Q\delta}{g}(v_1 - v)\, v \text{ ft.-lbs.} \tag{23}$$

To discover the value of v for which this will be a maximum, put

$\frac{dL}{dv} = 0$, that is, $v_1 - 2v = 0$, from which $v = \frac{v_1}{2}$. The expression for maximum theoretical power then becomes equal to

$$L = \frac{2Q\delta}{g}\left(v_1 - \frac{v_1}{2}\right)\frac{v_1}{2} = \frac{Q\delta}{g}\frac{v_1^2}{2} \text{ ft.-lbs. per sec.} \tag{24}$$

459. Efficiency of the Impulse Wheel. — The total energy per second in the water issuing from the nozzle is evidently equal to the weight of water per second multiplied by its fall, $= Q\delta h$, in which h is the total head in feet acting on the nozzle; so that the expression for *ideal hydraulic efficiency* (= ratio of energy received to total energy available) may be written

$$E_h = \frac{\frac{Q\delta}{g}(v_1 - v)(1 + \cos\theta)v}{Q\delta h}$$

$$= \frac{(v_1 - v)(1 + \cos\theta)v}{gh}. \tag{25}$$

This efficiency is theoretically 100 per cent if $\theta = 0$ and $v = \frac{v_1}{2}$, for then, remembering that $v_1 = \sqrt{2gh}$

$$E_h = \frac{\frac{v_1^2}{2}}{gh} = \frac{v_1^2}{2gh} = 1. \tag{26}$$

In practice, of course, this efficiency is never reached, on account of losses due to imperfect guidance, friction, etc., and the fact that θ cannot well be made exactly zero. The *actual* or *gross efficiency* of the wheel and nozzle in any case is, if W is the foot-pounds of work done by the wheel per second, as determined by Prony brake or other means,

$$E_a = \frac{W}{Q\delta h}. \tag{27}$$

The actual efficiencies of these wheels range from 70 to 90 per cent.

460. Turbines. — Church defines a turbine in general as follows: A water motor consisting of a number of short curved pipes set in a ring attached rigidly to a shaft upon which it revolves, and receiving water at all parts of its circumference from the mouths of other and fixed pipes or passageways.

Broadly, all water turbines are divided into two classes:

Impulse Turbines and Reaction Turbines. — In the first class all of the available head is converted into velocity before the water strikes the vanes. The pressure in the water remains constant during the passage through the wheel; air must, therefore, have free access to the passages, and the latter must not be completely full of water. It is, therefore, necessary that this turbine discharge into free air. The power of this turbine is due to the change in kinetic energy in the water.

In the reaction turbine, only a part of the head is converted into velocity before the water enters the wheel, and both pressure and velocity act on the vane. The passages must, therefore, be kept full of water. For this reason this turbine is often placed below the level of back water. The power is due to changes in kinetic energy and changes in pressure.

Turbines are also classified according to the direction of flow of the water, and we distinguish:

(*a*) Radial outward-flow turbines, in which the water flows generally at right angles to the shaft and is directed away from the shaft. In this case, the guide blades are placed within the wheel and the latter receives the water at the inner circumference.

(*b*) Radial inward-flow turbines, in which the water flows at right angles to the shaft, but the guide vanes are placed around the outer circumference of the wheel, the water flowing toward the shaft.

(*c*) Axial- or parallel-flow turbines, in which the path of a particle of water lies practically on the surface of a cylinder whose axis is that of the shaft; that is, the distance of the particle from the center of the shaft is constant.

(*d*) Mixed-flow turbines, in which the path of a particle of water is changed from radial inward flow at entrance to axial as the particle passes through the wheel.

In order to aid in understanding the action of the reaction turbines (the principal one to be considered), two fundamental principles should first be dealt with: the action of a stream upon a rotating vane, and the general case of amount of power developed by a turbine.*

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* Hoskins' method of treatment is followed; see Hoskins, Textbook on Hydraulics.

461. Action of a Stream upon a Rotating Vane: General Case.— *AB*, Fig. 611, is a vane rotating about the center *C* with a uniform angular velocity ω. The vane receives a stream at *B* and discharges at *A*. Let the absolute velocity of the entering stream be v_1, the relative velocity (relative to the vane) be w_1, and the velocity of *B* about *C* be u_1. At *A*, let the relative velocity (tangent to the last vane element) be w_2, the velocity about *C* be u_2, and the absolute velocity (resultant of u_2 and w_2) be v_2.

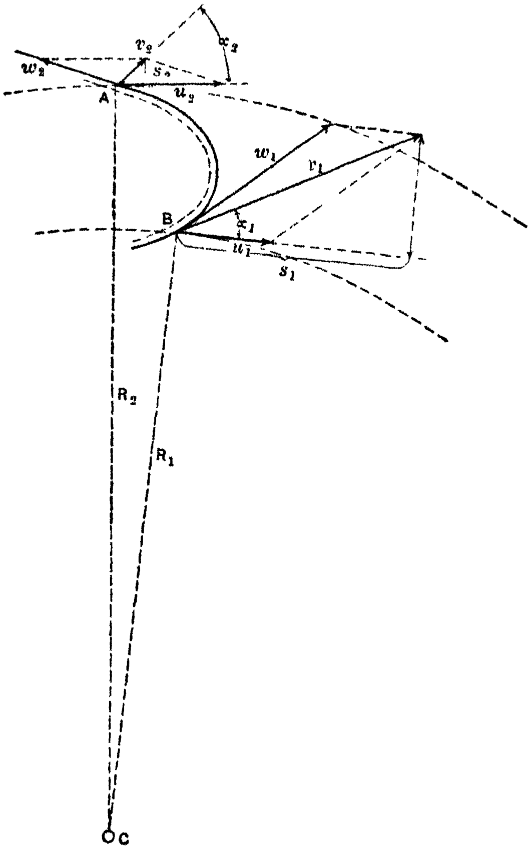

FIG. 611.

The angular momentum of a particle of water of mass ΔM at *B* is $\Delta M v_1 R_1 \cos \alpha_1$, in which R_1 is the radius *CB*, and α_1 is the angle between v_1 and u_1. Similarly, the angular momentum of the particle of mass ΔM at A is $\Delta M v_2 R_2 \cos \alpha_2$, in which R_2 is the radius CA and α_2 the angle between v_2 and u_2.

The action of the vane, therefore, causes a change in the angular momentum equal to

$$\Delta M\ (v_1 R_1 \cos \alpha_1 - v_2 R_2 \cos \alpha_2). \tag{28}$$

If W' is the weight of water striking the vane per second, the mass per second will be $\frac{W'}{g}$, and the change of angular momentum per second is

$$\frac{W'}{g}(v_1 R_1 \cos \alpha_1 - v_2 R_2 \cos \alpha_2). \tag{29}$$

It can be shown that this is also the *total moment** of the forces exerted by the water upon the vane. If there is a constant succession of these vanes, the weight W' becomes equal to the weight W furnished the wheel per second, and hence the

$$\text{Total moment} = \frac{W}{g}(v_1 R_1 \cos \alpha_1 - v_2 R_2 \cos \alpha_2). \tag{30}$$

Now, energy or work is equal to moment times angular velocity. So if we let ω = angular velocity, the energy received by the wheel in one second will be

$$L = \frac{W\omega}{g}(v_1 R_1 \cos \alpha_1 - v_2 R_2 \cos \alpha_2). \tag{31}$$

To simplify this expression for work, put $v_2 \cos \alpha_2 = s_2$, and $v_1 \cos \alpha_1 = s_1$. These are the components tangentially to the wheel of the absolute velocities v_1 and v_2 respectively, and are sometimes called velocities of whirl (see Fig. 611). Also, $R_2\omega = u_2$ = velocity of rotation of point A about center C, and $R_1\omega = u_1$. We may then write energy received per second

$$L = \frac{W}{g}(u_1 s_1 - u_2 s_2) \text{ ft.-lbs.} \tag{32}$$

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* With reference to this, Hoskins makes the following statement: Since the total increment of momentum of a particle per second is equal to the average value of the force acting upon the particle, it follows, by taking moments about any axis, that the total increment of the moment of momentum (= angular momentum) per second is equal to the average value of the moment of the force.

462. The Radial Outward-flow Reaction Turbine. — Fig. 612[*] shows a vertical cross section and a plan view of a Fourneyron turbine, which is the best-known example of this type An outline sketch of this turbine is shown in Fig. 613 The water from the head bay *H* enters the penstock *P* At the lower end of the penstock it enters the guide passages *gg*, in which its direction of motion is changed to the horizontal and with the direction of motion of the wheel *W* to avoid shock. The wheel passages are indicated at *g'g'*. The arrangement for controlling the quantity of water flowing, that is, the gate, is not shown in the sketch. Fig. 612, however, shows one method of doing it (gates at *CC*).

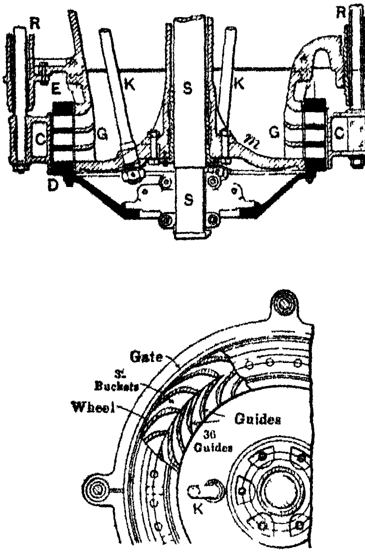

FIG. 612 — RADIAL OUTWARD FLOW REACTION TURBINE (FOURNEYRON)

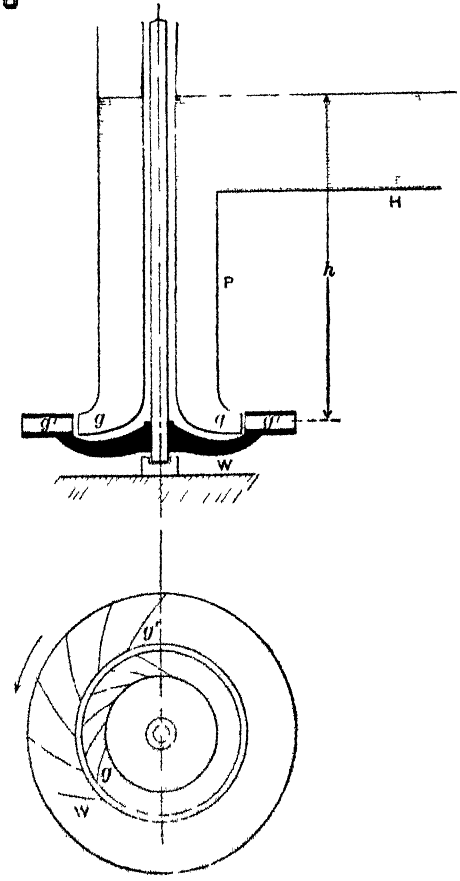

FIG. 613 — OUTLINE SKETCH, RADIAL OUTWARD-FLOW REACTION TURBINE.

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[*] Church, Hydraulic Motors, p. 111.

463. Power and Efficiency of the Radial Outward-flow Reaction Turbine. — Let Q be the cubic feet of water delivered to the wheel per second, δ = density of the water, and h be the total head from the level in the head bay to the level at which the turbine discharges (see Fig. 613), assuming that the discharge is into air.

Then the weight of water supplied per second is $Q\delta$ pounds, and the total available energy is $Q\delta h$ foot-pounds per second.

According to the previous paragraph, the actual energy received by the wheel from the water is

$$L = \frac{W}{g}(u_1 s_1 - u_2 s_2) \text{ ft.-lbs. per sec.,} \tag{33}$$

so that the actual hydraulic efficiency is

$$E_h = \frac{L}{Q\delta h} = \frac{\frac{W}{g}(u_1 s_1 - u_2 s_2)}{Wh} = \frac{u_1 s_1 - u_2 s_2}{gh}. \tag{34}$$

In this equation, the values of g, h, u_2, and u_1 are known, but s_1 and s_2 must be determined. The value of s_1 depends upon the absolute velocity v_1 (see Fig. 611) with which the water leaves the guide passage, which in turn is a function of the quantity Q of water flowing and the total cross section of the streams F_1 measured normal to the direction of v_1. s_2 depends upon v_2, which is the resultant of w_2 and u_2 (see Fig. 611). w_2 in turn depends upon Q and the total cross section F_2 of the streams leaving the wheel. F_2 is measured normal to the direction of w_2. Without entering into the derivation of the transformed equation, the final form is

$$\text{Hydraulic Efficiency } E_h = \frac{(\cos\alpha_1 - cc'\cos\alpha_2)\,v_1 u_1 - c^2 u_1^2}{gh}, \tag{35}$$

in which α_1 and α_2 are the angles indicated in Fig. 611, v_1 is the absolute velocity of the water entering the wheel, u_1 is the velocity of the inner end of the vane about the center of the wheel, $c = \frac{R_2}{R_1}$, and $c' = \frac{F_1}{F_2}$. All of these quantities may now be determined if Q and the dimensions of the wheel are known, by constructing a diagram like Fig. 611.

Equations (34) and (35) give the actual hydraulic efficiency of wheel and setting, because h is the total head from the level in the head bay to that in the tail race. Losses due to friction in penstock, to shock at wheel entrance, to kinetic energy in water leaving, etc., may each be expressed as an equivalent loss of head h', so that the net head will be $h - h'$. The efficiency equation may then be modified to express the actual efficiency of any part of the installation.

Equation (35) involves only turbine dimensions, wheel velocity u_1 and velocity v_1 of the water entering the runner. Since from any test v_1 may be found for any value of u_1, the actual hydraulic efficiency may be readily computed.

The *actual or gross efficiency* is, of course, the useful work done by the turbine, as determined by Prony brake or other means, divided by the available energy. If W_a represents the useful work in foot-pounds per second, so determined, the actual (sometimes called gross) efficiency will be

$$E_a = \frac{W_a}{Q\delta h}. \tag{36}$$

Actual efficiencies for the turbines range in practice from 80 to 90 per cent.

In the above development of the equations, the head h was taken to be the distance between the surface level in the head bay to the level at which the turbine discharged in air. Discharge into air is not the usual case. The turbine is either submerged, that is, placed below the level of the water in the tail race, or, if placed above the level, it discharges into a *draft* or *suction* tube.

In the case of the submerged wheel, the head h which is to be used for computing the available energy supplied to the wheel will be the distance from the surface of the water in the head bay to the level of the water in the back bay or tail race.

The use of a draft or suction tube is primarily for the purpose of making the turbine accessible, and it is found that if the tube is properly designed there will be no appreciable loss in efficiency. The end of the tube must be kept submerged and the tube must be air-tight. It need not necessarily be vertical. If h_1 is the height in feet from the discharge periphery of the wheel to the level of the water in the back bay, and 34 feet is taken as the height of the water barometer, the pressure, in feet, at the top of the draft tube will be $34 - h$ feet. The tube cannot be longer than 34 feet to the level of the back bay, or else the pressure at the top becomes negative and the tube will not remain full. In practice, 25 feet cannot usually be exceeded. As far as the effect upon the turbine is concerned, the use of the tube does not affect the head h to be used for computing available energy supplied. As before, this is equal to the distance between levels of head and back bays.

If the lower end of the draft tube have a gradually enlarging section, it can be shown that there is a saving in energy,* so that turbine efficiencies may be increased 2 or 3 per cent. Such an appliance is known as a *diffuser*.

464. Radial Inward-flow, Parallel-flow, and Mixed-flow Reaction Turbines. — One type of the radial inward-flow turbine is the Francis, another is the Thompson Vortex wheel. The best-known example of the parallel-flow is probably the Jonval, while most American turbines are mixed- (inward and downward) flow turbines.

Fig. 614† shows a Francis turbine. The wheel is supported by a collar bearing above the staging *S*. The casting *C* acts as a guide for the cylindrical gate *F* and also carries the wheel bearing *B*. The water flows as shown by the arrows, being first directed by the guide passages against the wheel vanes, flows radially through the wheel vanes, and then down and out of the turbine. The work, however, is done while the water is flowing radially. This type of turbine is sometimes called *center-vent* turbine. It may be submerged or discharge above the level of the back bay into a draft tube.

The Thompson vortex turbine differs from the foregoing in the method of controlling the quantity of water and in some details of guide- and wheel-blade design.‡

The principle of the Jonval turbine is illustrated in Fig. 615, and Fig. 616 shows a double vertical turbine of the type discharging into draft tubes.|| In Fig. 615 the water is received by the guide passages

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* See Church, Hydraulic Motors, p. 127.

† Lea, Hydraulics, p. 320.

‡ See Lea, Hydraulics, pp. 323 to 326.

|| Church, Hydraulic Motors, p. 123.

G and discharged against the wheel vanes G' in a direction v_1. The absolute path of a particle of water is shown by line AB, this path being at all points practically at a constant distance from the center of the turbine shaft.

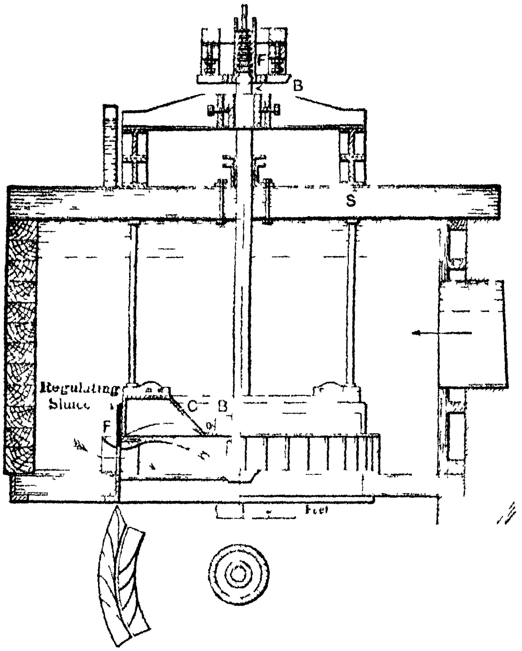

FIG. 614 —RADIAL INWARD FLOW REACTION TURBINE (FRANCIS)

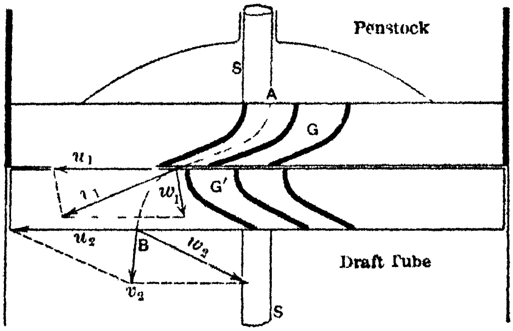

FIG. 615 —PRINCIPLE OF PARALLEL-FLOW REACTION TURBINE (JONVAL)

Figs. 617 and 618* illustrate the Risdon-Alcott cylinder-gate

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* From the catalogue of the company

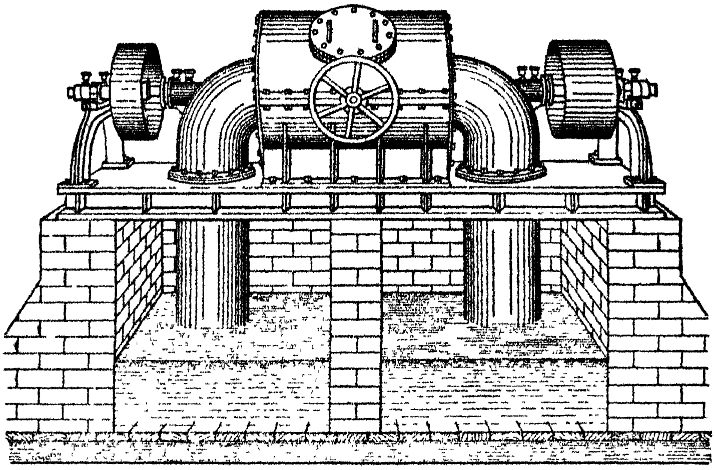

FIG. 616. — PAIR OF HORIZONTAL PARALLEL-FLOW REACTION TURBINES, WITH DRAFT TUBES (JONVAL).

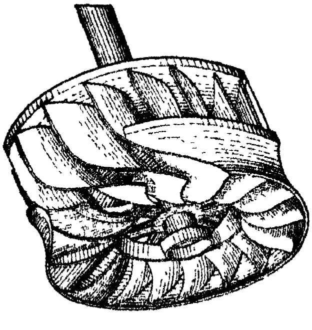

FIG. 617. — RUNNER OF A MIXED-FLOW REACTION TURBINE.

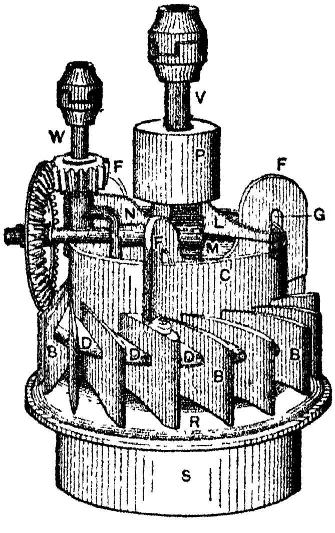

FIG. 618. — MIXED-FLOW REACTION TURBINE (RISDON-ALCOTT).

mixed-flow turbine. The wheel or runner clearly shows how the flow is changed from radially inward to vertically downward. In Fig. 618, *C* is the cylindrical gate which is moved up or down by the gearing shown, carrying with it the extension pieces *D* between the guide blades *B*. Pieces *D* form the upper limiting surface to the guide channels at any position of *C*. Guide blades *B* are fastened to the ring *R*.

465. Power and Hydraulic Efficiency of Radial Inward-flow, Parallel-flow, and Mixed-flow Reaction Turbines. — It would be beyond the scope of this book to enter into any exhaustive discussion of the theory of these turbines. For this the student must be referred to books on Hydraulics and Hydraulic Motors. The main concern here is to establish theoretical efficiency standards for comparison with actual efficiencies obtained on tests.

Velocity diagrams may be drawn for entrance and exit of each wheel, similar to the one outlined in Fig. 611. It will be found, after going through a demonstration like that in Arts. 461 and 463, that the final formulas there developed generally apply also to the case of the other wheels, that is, the equations for *power* and *hydraulic efficiency* are practically the same. To point out some of the changes and approximations in the use of the hydraulic efficiency formula, note first that $c = \frac{R_2}{R_1}$ is equal to 1.0 in the axial flow, and is less than 1.0 in the inward- and mixed-flow turbines, because R_1 is always the radius at the inlet and R_2 that at the outlet. Second, the formulas were derived on the assumption that R_1 and R_2 are constant. In the axial-flow turbine this is not quite true, but a fair approximation is possible. In the mixed-flow turbine, however, the outflow radius varies through a wide range and it is not easy to determine a fair value for R_2.

466. The Testing of Water Motors. — The objects of the test usually are the determination of efficiency and capacity at some constant speed of operation. Tests are run with various "gates," meaning different positions of the gates to control the admission of water, to determine variation of efficiency and power with variation in quantity of water supplied. Tests may also be made with varying heads of supply and at various speeds. There are also a number of special investigations that may be made.*

Tests for efficiency and capacity with various gate openings at one speed are often simply tests to determine whether manufacturers' guarantees are met. To bring out the full characteristics of a given wheel, however, a series of such tests at different speeds is desirable.

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* See the article by Schuster in the *Zeitschrift des Vereins deutscher Ingenieure* for methods of determining pressure and velocity conditions in the wheel of a Francis turbine.

The test of any type of water motor calls for the determination of the following items:

A. Volume or weight of water used in a given time.
B. The head through which the supply acts.
C. The power developed.
D. Speed of operation and gate opening (the latter of course applies only to turbines).

(*A*) *Volume or Weight of Water Supplied.*—The methods of determining the water supplied for a large wheel practically narrow down to a weir or current meter measurements in the tail race and to Pitot tube readings, if the conduit is closed. For the necessary precautions to be observed in these measurements and the method of computing the flow, see Chap. XII. See also the apron method of measurement in the same chapter. Measurement in head bays, if of regular construction, may be made by means of current meters and Pitot tubes.

For small wheels, the water supplied may often be measured at the inlet as well as at the outlet. Small flows may be caught in tanks and actually weighed. Somewhat larger flows may be determined in weir boxes. This is a good method in laboratory investigations. If the water is delivered to the wheel in a closed pipe, water meters, Venturi tubes, and Pitot tubes (see Chap. XII) become available. If the water is supplied through a nozzle, as in the Pelton or Doble wheels, the discharge from the nozzle (= water supplied to the wheel) may be found from the discharge coefficient of the nozzle and the pressure on the nozzle.

For turbines which are set in a penstock, the leakage of the penstock should be determined and allowed for, if the measurement of water supply is made by some means in the head bay.

(*B*) *The Supply Head.* — This may be determined either directly by measuring the vertical distance between surfaces of water in head and back bay, or indirectly by means of pressure gauges.

The first method is used where the heads are not great and where a datum plane of measurement may be easily established. The head so measured should be corrected for loss of head at entrance to penstock and loss due to friction in penstock, to give the effective head h on the turbine. These losses are not properly chargeable to the turbine as a machine. In properly designed installations these losses should be small.

When the water enters the turbine through a pipe, a pressure gauge may be placed in the pipe near the turbine to register the pressure head. To this pressure head, reduced to feet, must be added the velocity head in the pipe at the point where the static pressure is measured. The sum gives the head on the turbine at that point. To obtain the effective head h, add to this the vertical distance from the center of the gauge to the level of the water in the tail race. That is, if p is the pressure registered by the gauge, v is the velocity in the pipe as computed from quantity flowing and cross section, and h' is the distance from gauge to tail race, the effective head will be

$$h = \left(\frac{144\,p}{\delta} + \frac{v_2}{2\,g} + h'\right)\text{ft.}, \qquad (37)$$

in which δ = weight of one cubic foot of water.

In impulse wheels, like the Pelton or Doble, the head h is determined by the same equation, except that here h' is the distance in feet from the center of the gauge to the center of the nozzle.

In either case the sum of $\left(\frac{144\,p}{\delta} + \frac{v_2}{2\,g}\right)$ might have been found directly by using an impact tube, instead of static pressure tube, to determine the total pressure in the pipe.

(*C*) *The Power Developed.* — Equations for theoretical power developed have been given in previous articles. In the actual case the power developed is best determined by some form of absorption dynamometer, like the Prony or fluid friction brake (see Chap. X). If the turbine is direct-connected to an electric generator, the output of the generator corrected for efficiency may be taken to be the power developed by the wheel. Where neither of these methods can be used, and a sufficiently long length of shaft is available, some type of torsion meter (see Chap. X) may possibly find application.

(*D*) *Speed and Gate Opening.* — In high-speed wheels, the revolutions should be determined by some form of indicating or recording counter. For slow-speed wheels the ordinary counter may be used, although, since the power developed is a direct function of the speed, it is desirable in any case to have recording devices in order to determine the revolutions as accurately as possible.

In turbines it is necessary to have the gate-control arrangements calibrated or properly marked so that the ratio of the " part gate " to "full gate " can be readily set and determined during the tests.

The necessary *length of a " run "* in a series of tests depends only upon the length of time required to accurately determine the flow of water after the conditions for the run have been adjusted and have become constant. This length of time of course differs with the arrangements made for measuring the flow.

467. Report and Computations. - The form on page 1066 shows the observations necessary for a test of a water motor, together with the principal computations. It is assumed that a Prony brake is used to determine power developed, and that the flow is measured by a weir in the tail water.

Make runs as follows:

(*A*) With constant speed and a given head, vary the load from light loads to overloads, using, say, six different loads to obtain sufficient data for graphical representations. Compute the items called for on the blank.

(*B*) Repeat at the same loads for a number of speeds in a series from the lowest to the highest practicable, keeping the head constant.

(*C*) If it is possible to change the head, vary it by a series of steps and for each step repeat series *A* and *B*.

The form needs little explanation, the meaning of most of the quantities having been given in previous articles. For the computation of "total effective head," see Art. 466. The actual or gross efficiency is the ratio of D.H.P. (reduced to foot-pounds) to the available energy *Wh*. The ratio of velocity of periphery of wheel to velocity due to the head *h* is, in the case of the impulse wheel, computed to show how nearly the wheel meets the requirement that for maximum efficiency the ratio of these speeds is about 0.5.

TEST OF WATER MOTOR.

Type of Motor. .... Date.............................191

Principal Dimensions of Motor..

....

Details of Brake: Length of Arm, ft.. ....; Brake Zero, lbs. ....

Details of Weir: Type.................; Length, ft. ....; Weir Zero (Hook Gauge)...............

Observers: . . . . , ... , ... , ...

Ratio of Part Gate to Full Gate (for a Turbine).	Gauge on Supply Pipe, lbs. per sq. in.	Total Effective Head on Motor, ft (= h).	Hook-gauge Reading.	Depth of Water over Weir, ft	Water Used.		Revolution Counter.	Revolutions per Minute.	Reading of Brake Scale, lbs	Net Brake Load, lbs.	Developed Horse Power (= D.H.P)	Total Available Energy per Minute, ft.-lbs. (= Wh).	Actual or Gross Efficiency, per cent	Ratio of Velocity of Periphery of Wheel to Velocity Due to Head (for Impulse Wheel).
					Cubic Feet per Second (= Q).	Pounds per Second (= W).								

In the report compute, for the conditions for which best actual efficiency was found, the theoretical power and the hydraulic efficiency. (See Arts. 458 and 459 for these computations for the impulse wheel and Art. 463 for the equations for the reaction turbine.)

Represent the results of the tests graphically by drawing the following curves:

For a turbine, for a given head and constant speed, plot as abscissæ ratios of "part gate" to "full gate," and as ordinates (*a*) horse power developed, (*b*) actual efficiency, (*c*) quantity of water per second.

For an impulse wheel, for a given head and constant speed, plot as abscissæ quantity of water supplied per minute or per second, and as ordinates (*a*) horse power developed, (*b*) actual efficiency.

If either type of water motor were tested at other speeds, plot these curves for each speed.

Include in the report, if possible, drawings to scale to show the setting of the motor, the principal dimensions of motor, penstock or supply pipe, discharge tube, etc., and location and arrangement of testing apparatus.

PUMPS.

468. Reciprocating Pumps.-- These are usually steam-driven, and in that case their testing is considered under the head of *pumping engines* in Chap. XVIII, to which the student is referred. Where the prime mover is an electric motor, or the pump is operated by belt, the arrangements for testing the water ends remain the same. The input to the electric motor multiplied by the motor efficiency gives the power delivered to the pump. Where a pump is belt-driven some type of transmission dynamometer if one can be applied, (see Chap. X) is the best means for determining input.

469. Rotary Pumps. — These pumps are often confused with centrifugal pumps, but they are distinctly different (see next article), although, of course, the centrifugal pump is a rotary pump in the broad sense.

There are a number of rotary pumps on the market, differing mainly in construction of impeller. One type is that exemplified by the rotary blower, Fig. 562, p. 942. The advantage of these pumps seems to be that they possess no valves. On the other hand, it is difficult to keep them tight along the surfaces in sliding or rolling contact, so that the slip is very high against medium or light heads. The efficiency is correspondingly low.

The pumps may be operated by any type of prime mover, but are very often belt-driven. Their testing does not differ essentially from that of centrifugal pumps.

470. Centrifugal and Turbine Pumps. — These pumps are closely related to turbines; in fact, the radial outward-flow turbine could be made to pump water, if it were operated in a reverse direction by an external source of power. It will consequently be found that the equations developed for this turbine very nearly apply to the centrifugal pump, but it is perhaps better to develop a separate theory for the pump.

471. The Action of the Centrifugal Pump. — The following explanation of the action of a simple centrifugal pump follows mainly that of Hoskins.* No attempt will be made to discuss shapes of guide or impeller vanes, or proper angles of entrance or exit, the intention being merely to show how such a pump succeeds in lifting water.

If a simple vertical cylindrical vessel, *A* (Fig. 619), partly filled with water, is rotated about its vertical axis *XY*, the water soon takes up the motion of the cylindrical shell, and the two move as one body. It will be noted, however, that the surface of the water takes on a curved form, rising at the circumference and falling at the center. At any one speed of rotation the rise and fall will be a definite amount above and below the original level. It can be shown mathematically that the surface is a paraboloid of revolution, and that the distances *DE* and *DF* are equal. Let the head of water over the bottom at the point *E* be $= h$, then the head at any other point, such as *G*, will be $= h + y$. This extra head y is caused by the velocity c with which the particle of water at *G* revolves around the axis

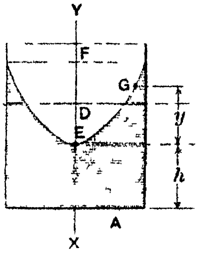

Fig. 619.

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* Hoskins, Textbook on Hydraulics.

XY, its position being due to the energy of rotation. The potential energy of the particle G of weight W at a distance y above the low point E is evidently $= Wy$, and this must be equal to the kinetic energy $\frac{W}{g}\cdot\frac{c^2}{2}$. From this $y = \frac{c^2}{2g}$, that is, y is the head corresponding to the velocity c of rotation.

Next take a closed vessel A of the shape shown in Fig. 620, assume that this vessel is completely filled with water, and that there is in the vessel a paddle wheel B rotating with uniform velocity about the axis XY. Then the body of water will take up the motion of the wheel, revolving with it (all effects of friction neglected). From the statement of the previous paragraph, there should then be set up pressures in the fluid varying from center to circumference as the square of the speed, and if piezometer columns C and D are inserted as shown, the liquid will rise to the heights indicated, that at C being equal to $\frac{c_1^2}{2g}$, that at D to $\frac{c_2^2}{2g}$, where c_1 and c_2 are the velocities indicated. The pressure head at D, therefore, exceeds

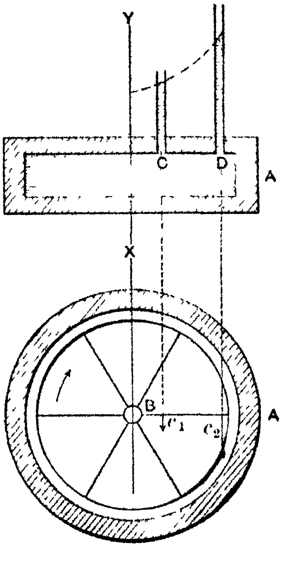

FIG. 620

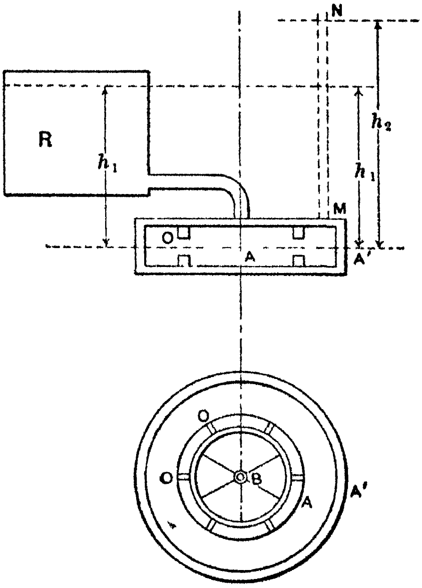

FIG. 621.

that at C by $\left(\frac{c_2^2}{2g} - \frac{c_1^2}{2g}\right)$ feet. This is the fundamental principle upon which the centrifugal pump operates.

To make the application of this to the pump a little clearer, consider Fig. 621. Here the wheel B revolves in the case A which is surrounded by an outer case A'. A communicates with A' through a number of passages OO. Water is furnished from a reservoir R under a head h_1. If the wheel B stood still, the water would rise in the pipe MN to the same height h_1 above the center of the wheel. But if the wheel is now rotated by external power, the pressure in the outer vessel A' will rise in the delivery pipe MN to some greater height h_2.

The strong resemblance of even this crude construction to the actual centrifugal pump is very striking. See Fig. 622,* which shows the construction of a horizontal centrifugal pump. The wheel B, rotating in the chamber A, is formed of an upper and lower sheet, the space between which is divided into a number of passages by vanes, corresponding to the arms of the paddle wheel. These vanes are not radial, like paddles, but generally of a curved shape, as shown, because that form is more efficient. Neither do they extend quite to the center, but leave a central opening B' for the suction water to enter. The water pumped enters the chamber A', from which it flows into the discharge pipe MN. The water is raised from the level of the supply by suction through a height h_s and is delivered through a height h_d, so that the total lift is $h_s + h_d$.

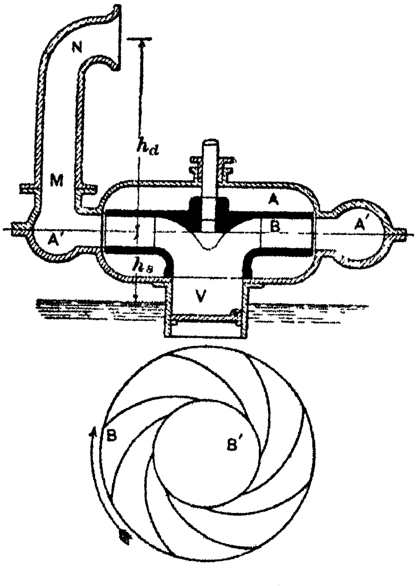

FIG 622.—HORIZONTAL CENTRIFUGAL PUMP.

A centrifugal pump cannot commence to discharge unless the wheel, casing, and suction pipe are full of water. For this reason, either a foot valve *V* must be provided in the suction pipe, or arrangements must be made so that the pump may be primed after it is started up.

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* From Lea, Hydraulics, p. 393.

The modern centrifugal pump is usually run upon a horizontal axis. The various constructions, by different makers, differ mainly with respect to shape of impeller passages, and means employed to convert the high velocities of the water leaving the impeller into useful pressure head, thus raising the efficiency of the pump. The following classification and discussion of characteristics is due to Prof. G. F. Blessing.[*]

I. *Impulse Pumps without Volute* (Fig. 623). — This is the cheapest and least efficient form. The impeller is a close fit in the concentric casing, and discharges the water at high velocity into the casing, in which there is no provision made for reducing the high velocity. These pumps are not much better than simple rotary pumps, because the water can be discharged from each passage for only about one-quarter of a turn. They are used mostly against small heads, — 2 to 6 feet, — and show efficiencies ranging from 30 to 45 per cent. They are largely used in dredging or irrigating operations, and show good resistance against shock or wear caused by solid material carried by the water.

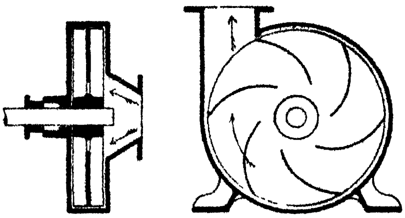

FIG. 623. — IMPULSE PUMP WITHOUT VOLUTE.

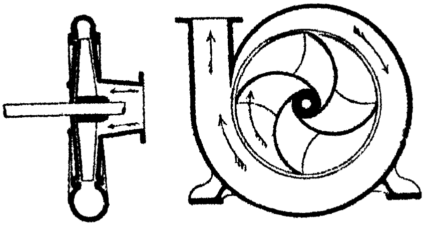

FIG. 624. — PUMP WITH VOLUTE, NO WHIRLPOOL CHAMBER.

II. *Pumps with Volute, but without Whirlpool Chamber* (Fig. 624). — The construction differs from the foregoing mainly in that a spiral discharge pipe, the volute chamber, is added to the casing. The form of the volute is practically that of the spiral of Archimedes,

[*] *Sibley Journal*, April, 1908.

that is, its cross section increases from practically zero up to the full area of discharge in such a manner as to keep the velocity of water in the chamber practically constant while the wheel is discharging all around the circumference. As in class I, no special provision is made in this class to convert the kinetic energy of the water into static pressure. Some transformation does take place, — that of the tangential component of the velocity with which the water leaves the impeller, — but the radial component is lost. The advantage of this type over class I, however, exists in the fact that the impeller is continuously discharging all around the circumference. This type of wheel is used for heads from 5 to 50 feet, and may show efficiencies up to 60 per cent.

III. *Pumps with Volute and Whirlpool Chamber* (Fig. 625). — The whirlpool chamber is a passage of annular ring form which is interposed between the impeller and the volute chamber. The water discharges into this chamber (which is usually filled with vanes having the same direction as the stream lines) with high velocity of whirl, by which is meant the tangential component of the absolute velocity of the water leaving the impeller. The water continues rotation in the whirlpool chamber, changing velocity into pressure head in so doing. The increase in efficiency due to the whirlpool chamber depends upon the construction of the latter.

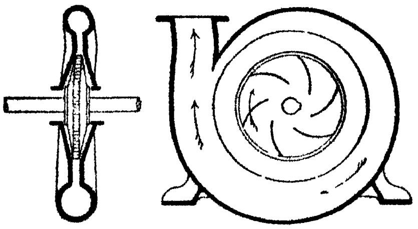

FIG. 625. — PUMP WITH VOLUTE AND WHIRLPOOL CHAMBER.

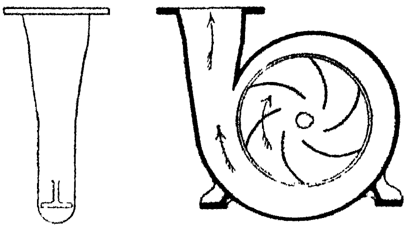

FIG. 626 — PUMP WITH VOLUTE AND BELL-MOUTH DISCHARGE.

IV. *Pumps with Volute and Bell-mouth Discharge* (Fig. 626). — These pumps are the same as class II, except that the discharge pipe gradually enlarges, to convert velocity into pressure head. This construction is of importance for pumps of low lifts; for those discharging against high heads it does not add much to the efficiency. When the velocity of water in the volute chamber is about

15 feet per second (common figure), the gain in head due to the bell-mouth discharge is usually less than 3 feet, which may, of course, be a large percentage for a low lift, but a small percentage for a high head.

V. *Single-stage Turbine Pump* (Fig. 627). — The special feature of turbine pumps is that the impeller discharges into a number of expanding nozzles, which in turn discharge into a concentric case. The ring containing these nozzles is called the *diffuser,* and the function of the nozzles is to convert kinetic energy into pressure head before the water reaches the discharge chamber. Their function is thus the same as that of the whirlpool chamber, but the latter is not suited to great reductions of velocity unless it be made very deep, which would make the pump very bulky. The turbine pump is used as a single-stage pump for heads from 50 to 150 feet and may give efficiencies exceeding 80 per cent.

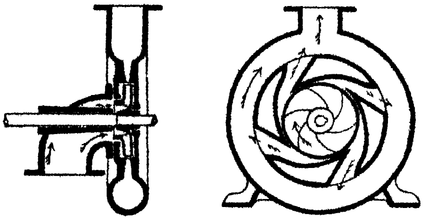

FIG. 627. — SINGLE-STAGE TURBINE PUMP

VI. *Multistage Turbine Pump* (Fig. 628). — The single-stage turbine pump, if used against a high head, requires high velocities of rotation, which causes excessive skin friction between metal and water. To overcome this, the pump is compounded by mounting several impellers on the same shaft. Each impeller operates in a

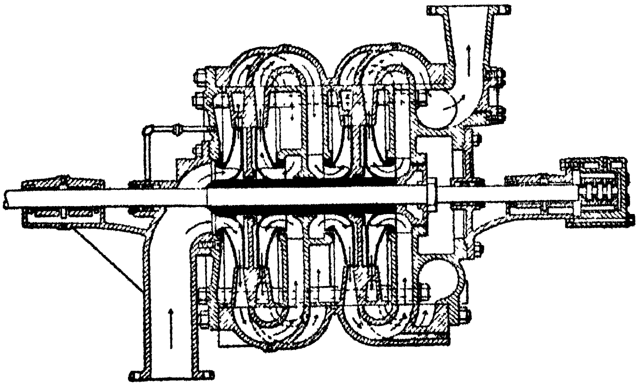

FIG. 628. — MULTISTAGE TURBINE PUMP.

separate chamber and forms a complete pump in itself, the first delivering the water to the suction of the second, etc. Multi-stage pumps with 8 stages delivering against 2000-foot head have been built.

472. Theoretical Power, Manometric, Hydraulic, and Actual Efficiencies of Centrifugal Pumps. — If the action of a pump impeller is analyzed in the same way as was done for a turbine runner of the radial-outflow type, which, as stated, is an exactly parallel case, a formula of the same form as equation (32) may be developed for the work done upon the water. This equation is

$$L = \frac{W}{g}(u_1 s_1 - u_2 s_2) \text{ ft.-lbs. per sec.} \tag{38}$$

The meaning of u and s may be read from Fig. 611. In the case of the turbine runner, the velocity u_1 is higher than u_2, and L is positive, meaning that the water does work upon the runner. In the case of the pump impeller, u_2 is greater than u_1, hence L is negative, which means that work is done upon the water. To avoid the use of the negative sign, we may write *theoretical work done upon the water*

$$= L = \frac{W}{g}(u_2 s_2 - u_1 s_1) \text{ ft.-lbs. per sec.} \tag{39}$$

The water enters the impeller practically radially, so that $s_1 = v_1 \cos \alpha_1 = 0$ (see Fig. 611). We also have $s_2 = u_2 + w_2 \cos \alpha_2$. The theoretical work then is

$$L = \frac{W}{g} u_2 (u_2 + w_2 \cos \alpha_2) \text{ ft.-lbs. per sec.,} \tag{40}$$

in which u_2 = tangential velocity at exit from impeller passage;
w_2 = velocity of water relative to vane;
α_2 = angle between u_2 and w_2 (see Fig. 611).

w_2 may be computed from the quantity Q of water flowing per second and α_2 is known from wheel dimensions.

If the theoretical work done is divided by the weight W of water flowing, the result will be the *theoretical total lift or head.* This then is equal to

$$H = \frac{u_2 (u_2 + w_2 \cos \alpha_2)}{g} \text{ ft.} \tag{41}$$

The *actual gross head* through which a pump operates is in practice found as follows:

Connect a manometer to the suction pipe of the pump close to the latter, and let the reading of this manometer, when the pump is in proper operation, be equivalent to a static head of h_s feet, measured above atmosphere. (The reading will usually be inches of mercury, which must be reduced to the equivalent water column h_s in feet.*)

Connect a pressure gauge to the discharge main close to the pump, and let the reading of the gauge be equivalent to a discharge head of h_d feet, measured above atmosphere. (The reading will usually be in pounds per square inch.)

Let the velocity of the water through the suction pipe, at the point where h_s is measured, be v_s feet per second, and the velocity through the discharge main at the point where h_d is measured be v_d feet per second. These velocities are computed from the quantity of water flowing and the dimensions of the mains.

Let the water barometer (atmospheric pressure) = h_b feet (about 34 feet).

Now, on the basis of Bernouilli's theorem, write the energy equations for the two points. Neglect friction between the points, for that is directly chargeable to the pump, if the pressure-measuring instruments are connected very close to the pump casing, as is assumed to be the case. If W is the quantity of water flowing per second, and h the gross head in feet produced by the pump, the energy imparted to the water in passing through the pump is evidently = Wh foot-pounds. Express the total pressures acting at the two points of measurement as absolute pressures, and let the vertical distance between the point of manometer connection to the center of the discharge gauge be h_1 feet. Then the energy equation will read

$$W \overbrace{\left\{ \overbrace{h_b + (\pm h_s)}^{\text{Static Head}} + \overbrace{\frac{v_s^2}{2\,g}}^{\text{Velocity Head}} \right\}}^{\text{Suction Side}} = W \overbrace{\left(\overbrace{h_b + h_d + h_1}^{\text{Static Head}} + \overbrace{\frac{v_d^2}{2\,g}}^{\text{Velocity Head}} \right)}^{\text{Discharge Side}} - \overbrace{Wh}^{\text{Energy Input}} \tag{42}$$

---

* h_s may be intrinsically positive or negative. If the pump is supplied with water under some pressure, then h_s is intrinsically positive, and in the equation must be written = $(+ h_s)$. If, on the other hand, the pump lifts the water, h_s is intrinsically negative, and must be written = $(- h_s)$.

Now, assuming that the pump lifts water, so that h_s is intrinsically negative, we have, canceling W,

$$h_b + (-h_s) + \frac{v_s^2}{2g} = h_b + h_d + h_1 + \frac{v_d^2}{2g} - h,$$

from which the gross head through which the pump operates is

$$h = \left(h_d + h_s + h_1 + \frac{v_d^2 - v_s^2}{2g}\right) \text{ft.} \tag{43}$$

Similarly, if the pump receives water under pressure higher than atmospheric so that h_s is intrinsically positive, we have

$$h = \left(h_d - h_s + h_1 + \frac{v_d^2 - v_s^2}{2g}\right) \text{ft.} \tag{44}$$

It should be noted here that h_d and h_s are not simply the actual vertical distances from the point where the gauge is connected to the discharge level, or from the point where the manometer is connected down to the suction level. In each case h_d and h_s include besides this the friction loss in the pipe. To find h_d and h_s simply by measurement of the actual distances assumes negligible friction losses in the piping.

The *useful* head pumped through is equal to

$$h_u = (h_d' + h_s' + h_1) \text{ ft.,} \tag{45}$$

where h_d' is the actual lift from center of gauge to discharge level and h_s' is the actual suction lift in feet.

Manometric or Hydraulic Efficiency of a centrifugal pump is defined as the ratio of the actual gross head pumped through divided by the theoretical total head; that is,

$$E_{\text{man.}} = \frac{h}{H} = \frac{h_d + h_s + h_1 + \frac{v_d^2 - v_s^2}{2g}}{\frac{u_2(u_2 + w_2 \cos \alpha_2)}{g}}$$

$$= \frac{g\left(h_d + h_s + h_1 + \frac{v_d^2 - v_s^2}{2g}\right)}{u_2(u_2 + w_2 \cos \alpha_2)}. \tag{46}$$

The *hydraulic efficiency* is by some writers computed on another, though equivalent, basis and is defined as the ratio of the total work done by the pump to the power input to the wheel. The latter is not the theoretical input L of equation (40), but the total input to the pump less the power lost in axle friction. Let L_1 be the actual power input in foot-pounds per second, and L_2 be the power lost in bearing friction. The input to the wheel is $L_3 = L_1 - L_2$, and, by definition, hydraulic efficiency then is

$$E_h = \frac{Wh}{L_3} = \frac{W\left(h_d + h_s + h_1 + \frac{v_d^2 - v_s^2}{2g}\right)}{L_3}. \tag{47}$$

The actual efficiency is of course always determined by the ratio of total work done by the pump to the total energy input, so that

$$E_a = \frac{Wh}{L_1} = \frac{W\left(h_d + h_s + h_1 + \frac{v_d^2 - v_s^2}{2g}\right)}{L_1}. \tag{48}$$

473. The Testing of Rotary and Centrifugal Pumps. — The test is usually for power input, capacity, and efficiency. The determinations and observations required are:

(*A*) Power input;
(*B*) Quantity of water pumped;
(*C*) Total and useful heads pumped against;
(*D*) Power output (foot-pounds per minute or horse power);
(*E*) Speed.

(*A*) Power input is determined by any of the means already described if the prime mover is direct-connected (steam engine or turbine, electric motor, etc.). When belt-driven a transmission dynamometer should be used.

(*B*) The quantity of water pumped may be determined either on the suction or discharge sides by Pitot tubes, by common water meter or Venturi meter on the discharge side, or by nozzles, orifices, or weirs, as may be convenient (see Chap. XII for available methods).

(*C*) Manometers and gauge are connected to suction and discharge mains and readings taken, as outlined in previous paragraph. The suction and discharge heads may be varied at will, if desired, by placing throttle valves in the suction and discharge mains, the former below the measuring instruments in the suction pipe, the latter beyond the instruments in the discharge pipe.

(*D*) The power output is equal to Wh foot-pounds per second, if W = water pumped per second and h is total head in feet. Horse power is then $= \frac{Wh}{550}$.

(*E*) Speed should be taken by some recording form of counter.

As in the turbine, the *length of each test* is determined mainly by the time necessary to make sure of the rate of flow of water, after conditions have become constant.

The following forms are used at Sibley College in the testing of a small centrifugal pump. The pump is belt-driven by a direct-current motor, and a weir is used to measure the discharge. In view of the explanations given in previous articles of this chapter, none of the items need any further explanation, with the exception, perhaps, of the item "Duty." The B.t.u. supplied the pump is in this case obtained by converting the applied horse power or power input to equivalent heat units per hour by multiplying by 2545.

TEST OF CENTRIFUGAL PUMP.

Log Sheet.

Date of Test . . . . . 19 . Observers, ... , ...

Details of Weir: Type . . ; Length, ft. ; Hook-gauge Zero. . . . .

Number.	Time.	Reading of Manometer (in. Hg.).	Reading of Gauge (lbs. per sq in.)	Hook-gauge Reading.	R.P.M. of Motor.	R P M. of Pump.	Amperes.	Volts.	Watts.
....	......	....	....	....	....	....	....	......	. . . .
....	......	....	.....	. . . . .	....	....	. . ..	......	......
....	......	....	. . . . .	....	....	....	. . . .	..... .	.....
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....	.....	..	. . . . . .	. .	. . .	....	.. . . .	......	.. . .

TEST OF CENTRIFUGAL PUMP.

Result Sheet.

Type of Pump..

Details of Impeller: Diameter, inches...........; Width, inches..................

Diameter of Suction Pipe, inches............; of Discharge Pipe, inches............

Run Number.	1	2	3	4	5	6
Av. suction head, by manometer, ft., h_s						
Av. discharge head, by gauge, ft., h_d..						
Distance between manometer and gauge, ft., h_1....						
Velocity head in suction, ft., $\frac{v_s^2}{2g}$........						
Velocity head in discharge, ft., $\frac{v_d^2}{2g}$......						
Total head, feet, h................. ..						
Hook-gauge reading.............. ..						
Head on weir, feet....................						
Discharge, cu. ft. per sec., Q.......						
R.P.M. of motor............... ...						
R.P.M. of pump.						
Velocity of periphery of impeller.						
Velocity in discharge main, ft. per sec., v_d						
Velocity in suction main, ft. per sec., v_s....						
Amperes						
Volts...						
Input to motor, H.P.						
Efficiency of motor, per cent....... ..						
Input to pump, A.H.P						
Pounds of water discharge per sec., W .						
Actual efficiency of pump, per cent, E_a..						
Hydraulic efficiency, per cent, E_h ..						
Manometric efficiency, per cent, E_{man} . .						
Efficiency of set, motor included, per cent						
Capacity in gallons per min.... ...						
Duty in ft.-lbs. per 1,000,000 B.t.u. supplied......						

474. Scope of Tests and Report.—The following series of runs may be made on the pump:

(*A*) With speed and suction head constant, vary the discharge head by controlling the valve in the discharge main.

(*B*) With speed and discharge head constant, vary the suction head.

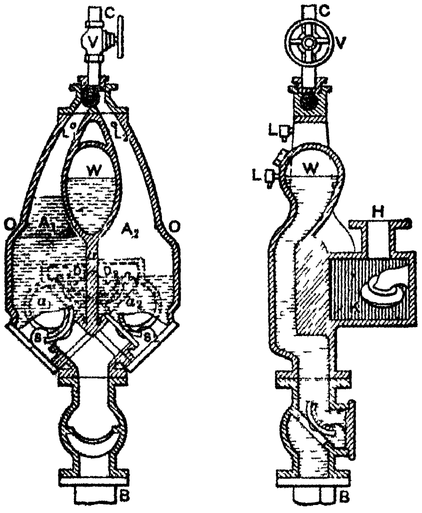
(*C*) Repeat series (*A*) and (*B*) for a series of different speeds.

To exhibit the characteristics of the pump, draw the following curves:

With capacity in gallons per minute as abscissas, use the following ordinates: (*a*) head *h* in feet; (*b*) useful horse power; and (*c*) actual efficiency.

If a series of speeds are used, one set of such curves should be drawn for each speed.

475. The Pulsometer. — This is a pump in which the direct pressure of steam upon water is used to drive the water out of a chamber and up a discharge pipe. Figs. 629 and 630[*] show two sections of a pulsometer.



FIGS. 629 AND 630. — PULSOMETER.

It consists of two chambers, A_1A_2, separated by a partition *a*. The chambers are connected to a common suction pipe *B* through suction valves S_1 and S_2. They communicate with a common discharge chamber *K* by discharge valves D_1 and D_2. Chamber *W* connects with the suction pipe through a side passage and serves as an air chamber. The ball valve *E* controls the course of the steam, the supply of which is regulated at *C*.

In the position shown, with *E* closing the left-hand opening, the water is rising in A_1, S_1 being open, while the pressure of the steam on top of the water in A_2 forces this out through D_2 into *K* and up *H*. The speed of operation in the two sides is so controlled that, as the water is all pushed out of A_2, the water in A_1 has risen nearly to the top, pushing any air that may be in A_1 ahead of itself against the valve *E*. At the next instant, steam starts to escape through D_2 and is rapidly condensing or mixing with the water in *K*. This causes the pressure in A_2 to fall to such an extent that the overpressure in A_1 pushes the valve *E* over to the right-hand seat. It is understood, of course, that the pressure in A_1, up to the instant the valve E shifts, is not up to atmosphere, because it is produced by water flowing into A_1 under the influence of atmospheric pressure, but the pressure difference between A_1 and A_2 is enough to throw E. The steam then does work in A_1 while A_2 is filling. The operation is then repeated.

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[*] Mechanics of Pumping Machinery, Weisbach & Hermann, p. 287.

The rapidity with which a chamber is emptied depends of course upon the steam pressure. It is essential for steady operation that the filling and emptying operation should take about the same length of time. To regulate this, each chamber is fitted with an air valve L opening inward, so set that at each operation a certain amount of air is taken in. This controls the pressure in the suction chamber and hence the time of filling. The rapidity of emptying is controlled by regulating the valve C.

The analysis of the operation of this instrument is very similar to that of the steam injector, except that the steam acts by pressure and not by impact.

Let w = weight of steam used per hour;
W = weight of water pumped per hour;
h_d = discharge head in feet;
h_s = suction head in feet;
h_1 = distance in feet between centers of suction and discharge gauges;
p and x = pressure per square inch and quality of steam in the main;
t_1 = temperature of suction water;
t_2 = temperature of discharge water.

Neglect changes in velocity heads, as the velocities are low in any case. Then, assuming that the loss of heat by radiation from the pulsometer wall is R heat units per hour, we may write the heat equation

$$w\,(xr + q - q_d) = W\,(t_2 - t_1) + R, \tag{49}$$

in which r and q are taken from the steam table for the pressure p, and q_d is the heat of the liquid corresponding to the temperature t_2. It is here assumed that the steam loses no heat in passing the admission valve. If it is possible to make a close approximation to the value of R, and W is determined, the amount of steam w used may be approximately computed from the equation. For an accurate test, of course, both W and w should be determined for which the injector set-up described in Chap. XX may be used.

The heat equivalent of the mechanical work done by the pulsometer per hour is

$$U = \frac{W(h_s + h_d + h_l) + wh_d}{778} \text{ B.t.u.} \qquad (50)$$

The heat expended in thermal units per hour is

$$H = w(xr + q - q_d) \text{ B.t.u.} \qquad (51)$$

Hence the pump efficiency is

$$E = \frac{U}{H} = \frac{W(h_s + h_d + h_l) + wh_d}{778\, w(xr + q - q_d)}. \qquad (52)$$

476. The Humphrey Internal-combustion Pump. — This pump uses the expansion of a combustible mixture of gases, after explosion, for the pumping of water. Fig. 631 shows a conventional sketch of the principal parts of a four-cycle pump.* Its operation is rather complex, but may be explained briefly as follows: C is a conical combustion chamber with inlet and exhaust valves at the top. There is also a scavenging valve, which is not shown. An interlocking gear is arranged so that either valve I or E is locked shut when the other is operating. Suppose that a charge of gas is compressed in the upper part of C, the water standing at some level in C, and let the charge be exploded by spark in the ordinary way.

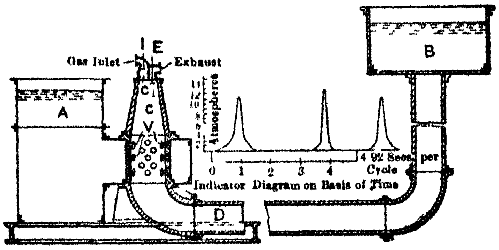

FIG 631. — HUMPHREY INTERNAL-COMBUSTION PUMP.

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* From an article on the Humphrey Internal-combustion Pump, by E. N. Trump, in the *American Machinist*, Jan. 5, 1911. This article also shows the two-cycle pump, applications of this type of pump to air compression, etc.

Valves *I* and *E* being shut, the pressure generated starts to depress the water column, forcing it toward the high-level tank *B*. The pressure falls as the water column attains velocity, so that by the time the gas pressure has reached atmospheric pressure the water column has considerable velocity. Since this motion cannot be suddenly arrested, expansion below the atmosphere in *C* takes place, which opens both the exhaust valve *E* and the water-suction valves *V*. The water coming in mostly follows the column of water moving towards the high-level tank, but the level will also rise in *C* in the effort to attain the level in the suction tank *A*. The kinetic energy of the water column is of course finally used up in forcing part of the water into the high-level tank. After this the water starts to surge back, filling the combustion chamber *C* and shutting the exhaust valve *E* by impact. This imprisons a quantity of burned gas in the space C_1, which gas, owing to the inertia of the water column, is compressed to a pressure higher than that of the static head due to the suction tank. A second outward surge of the water column in *C* soon results in decreasing the pressure in C_1 to below atmosphere, when the valve *I* opens against a light spring and a new combustible charge is drawn in. The next return motion of the water column compresses this charge, and at the moment of maximum compression the igniter operates, after which the cycle is repeated. The only moving parts of this pump are the valves, and they are all on their seats when the explosion occurs.

The principal measurements required in a test on this pump are, of course, the quantity of gas used and the amount of water pumped, together with the head pumped through. The latter would in the case shown in Fig. 631 be the difference between the levels of the discharge and suction tanks. A pump of somewhat different design from that shown in Fig. 631 has shown thermal efficiencies approximating 22 per cent. This pump had a straight cylinder 25 inches in diameter by 48 inches long, the capacity was about 4000 gallons per minute, and 35 water horse power were developed.

The Humphrey pump is also built on the two-cycle principle and has been adapted to the compression of air. It is already being built in large capacities, several four-cycle pumps, to have a capacity of 40,000,000 gallons in 24 hours, being under construction.

477. Pumping by Compressed Air: Pneumatic Pumps, Air Lifts.—There are two classes of pumps using compressed air for the raising of water. The first class uses the air expansively, and these pumps are generally called *air lifts*. The second uses the air practically without expansion. These pumps are known as *pneumatic displacement pumps*, and operate very nearly on the principle of the pulsometer.*

The principle of the air lift is very simple. The pump consists essentially of a delivery pipe let down into the well, and a smaller air pipe which connects to the discharge pipe somewhere near the foot. In the simplest type, the air pipe is carried alongside the delivery pipe, is turned upward and ends in a nozzle (see Fig. 632). This is the construction often used when the delivery pipe is comparatively small. In such a case the air does not mix with the water to any extent. On leaving the nozzle it expands to fill the delivery pipe and rises in this in masses, which may be compared to pistons, each separating two bodies of water. Where the delivery pipe is larger, it is better to admit the air through a ring of ports around the delivery pipe. The air then mixes with the water in small bodies or bubbles. The action is the same in either case; the water is raised mainly by the buoyancy of the air, or, as Peele states it, by the aëration of the water column which causes a reduction of the specific gravity.

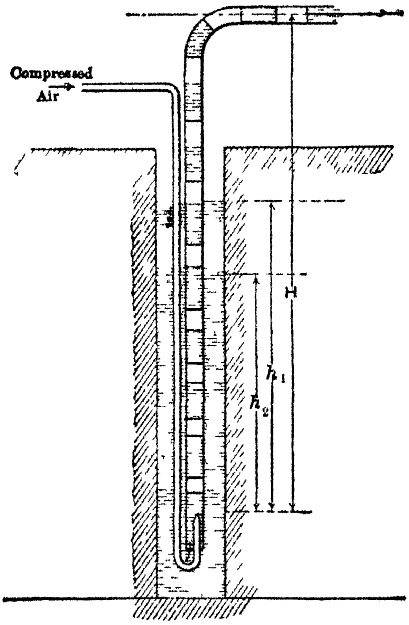

FIG. 632.—PUMPING BY COMPRESSED AIR. THE AIR LIFT.

In Fig. 632 let h_1 be the depth of submergence of the delivery pipe to the place at which the air enters, h_2 this distance when the well is in operation, and H be the total lift. The pressure of the air at the place of entrance is theoretically equal to the head of mixed water and air above it in the delivery pipe. As the air rises, the pressure decreases and the air expands, so that near the outlet the air pressure is but little above atmosphere. The initial pressure necessary is determined by the head of water h_1. If the submersion is too deep as compared with the net lift $H - h_2$, the pressure required is too high, the work required at the air compressor is necessarily too great, and the efficiency of the installation is correspondingly low. Too little submergence calls for a larger volume of air to produce the required velocity, and this again means low efficiency. The total efficiency under best conditions, from the foot-pounds of work done at the compressor to the foot-pounds in water delivered, may be over 50 per cent for low heads (10 to 30 feet); beyond this the efficiency rapidly falls with the head, decreasing to about 20 per cent with heads approximating 100 to 130 feet.

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* For a thorough discussion of air lifts and pneumatic pumps, see Greene, Pumping Machinery. See also Peele, Compressed-air Plant.

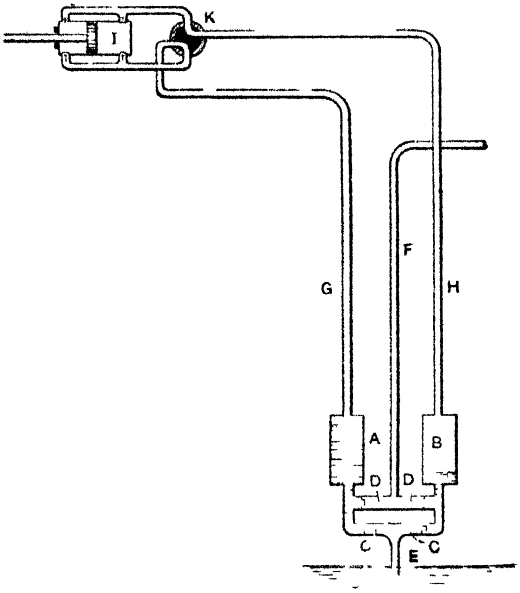

FIG. 633 — PUMPING BY COMPRESSED AIR. PNEUMATIC DISPLACEMENT PUMP.

A complete test of an air lift requires the determination of the following quantities: (*a*) air-compressor horse power; (*b*) volume of free air compressed; (*c*) quantity of water pumped; (*d*) net lift ($H - h_2$ in Fig. 632). The total efficiency is the ratio of the foot-pounds of work done in water lifted to the foot-pounds of work done in compressing the air.

The principle of the pneumatic displacement pump may be explained from Fig. 633. Two vessels *A* and *B* are connected to suction valves *CC* and discharge valves *DD*, as shown. *E* is the suction, *F* the discharge pipe. Air pipes *G* and *H* are connected to the compressor *I* through the compound valve *K* so that air may either be drawn from or forced into either vessel. In the sketch as shown, the compressor piston is moving to the right and is forcing air into cylinder *B*, forcing out the water, and drawing air out of the cylinder *A*, which fills the cylinder with water. At the end of the stroke, *B* is emptied of water and filled with air, while *A* is filled with water. Valve *K* is then thrown over to the other position and the operation is repeated in the reverse way. The height to which water can be lifted depends upon the air pressure used. The suction lift is of course governed by the same laws as in other pumps.

A test of a pump of this type requires the determination of substantially the same quantities as outlined for the air lift. The total efficiency is computed in the same way.

APPENDIX

APPENDIX.

LIST OF TABLES.

NO. PAGE
1. Coefficients of Friction and Angles of Repose 1090
2. Beaumé and Specific Gravity Scales 1091
3. Steam Tables.
    I. Temperature Table 1092
    II. Pressure Table 1095
4. Horse Power per Pound of Mean Effective Pressure 1099
5. Relative Humidity, 30-in. Barometer 1100
6. Theoretical Water Rates for Steam Engines (Thompson's Table) 1105
7. Properties of Ammonia 1107
8. Properties of Sulphur Dioxide 1108
9. Properties of Carbon Dioxide 1108
10. Absorption of Ammonia in Water 1109
11. Properties of NaCl Brine 1109
12. Properties of $CaCl_2$ Brine 1110

This list is followed by a curve from which the weight of water in pounds per cubic foot may be found for temperatures from 20° to 360° Fahr.

Other tables in the body of the book are·

Kinds of Tests required on Various Structural Materials 138
Relation between Pressure in Pounds per Square Inch and in Inches of Mercury and Feet of Water 167
Melting Points and Specific Heats of Metals 207
Useful Formulæ for Uniform Motion 238
Specific Gravity of Oils 247
Density, Specific Heat, etc., of Gases 328
Coefficients of Discharge for Water, for Circular, Square, and Rectangular Vertical Orifices 363
Coefficients of Discharge for Air, for Circular Orifices 425
Weights and Volumes of the Principal Substances Concerned in Combustion 466
Heating Value of Principal Combustible Elements 467
Mean Specific Heat at Constant Pressure for Principal Gases Concerned in Combustion 469
Limits of the Throttling Calorimeter 564
Atomic Weights, Molecular Weights, Density, Specific Volume, etc., of the Principal Gases 857
Heating Value, Air Required, and Products of Combustion for Hydrogen and Principal Hydrocarbons 858
Average Composition of Principal Fuel Gases 861
Principal Data for the Most Important Refrigerants:
    Ammonia, Carbon Dioxide, and Sulphur Dioxide 995
Coefficients of Heat Transmission for Various Insulating Materials 1011

TABLE 1. — COEFFICIENTS OF FRICTION (MORIN).

(Chap. IX, p. 236)

No	Surfaces.	Angle of Repose.	Coefficient of Friction	
		ϕ	$f = \tan \phi$	$1 + f$
		Degrees		
1	Wood on wood, dry.	14 to 26½	0 25 to 5	4 to 2
2	Wood on wood, soaked	11½ to 2	2 to 04	5 to 25
3	Metals on oak, dry.	26½ to 31	5 to 6	2 to 1 67
4	Metals on oak, wet.	13½ to 14½	24 to .26	4.17 to 3 85
5	Metals on oak, soapy	11½	2	5
6	Metals on elm, dry.	11½ to 14	.2 to 25	5 to 4
7	Hemp on oak, dry . . .	28	53	1 89
8	Hemp on oak, wet	18½	33	3
9	Leather on oak	15 to 19½	27 to 38	3 7 to 2 86
10	Leather on metals, dry . . .	29½	56	1 79
11	Leather on metals, wet. . . .	20	.36	2 78
12	Leather on metals, greasy. . .	13	23	4 35
13	Leather on metals, oily. . . .	8½	15	6 67
14	Metals on metals, dry	8½ to 11½	15 to 2	6 67 to 5
15	Metals on metals, wet	16½	3	3 33
16	Smooth surfaces, occasionally greased	4 to 4½	.07 to 08	14 3 to 12.
17	Smooth surfaces, continually greased	3	05	20
18	Smooth surfaces, best results. . . .	1¾ to 2	03 to 036	33 3 to 27 0
19	Bronze on lignum vitæ, wet . . .	3?	.05?	20?

Note — The above table is defective, since the pressure per square inch is not given The coefficient of friction diminishes with increase of pressure, so that in some cases the total friction remains constant.

TABLE 2. — TABLE OF BEAUMÉ'S HYDROMETER SCALE WITH CORRESPONDING SPECIFIC GRAVITIES.

(Chap. IX, p. 246.)

FOR LIQUIDS LIGHTER THAN WATER. TEMP. 60° FAHR.

Beaumé.	Specific Gravity.	Beaumé.	Specific Gravity.	Beaumé.	Specific Gravity.	Beaumé.	Specific Gravity.
10	1.0000	31	0.8695	52	0.7692	73	0.6896
11	0.9929	32	0.8641	53	0.7650	74	0.6863
12	0.9859	33	0.8588	54	0.7608	75	0.6829
13	0.9790	34	0.8536	55	0.7567	76	0.6796
14	0.9722	35	0.8484	56	0.7526	77	0.6763
15	0.9655	36	0.8433	57	0.7486	78	0.6730
16	0.9589	37	0.8383	58	0.7446	79	0.6698
17	0.9523	38	0.8333	59	0.7407	80	0.6666
18	0.9459	39	0.8284	60	0.7368	81	0.6635
19	0.9395	40	0.8235	61	0.7329	82	0.6604
20	0.9333	41	0.8187	62	0.7290	83	0.6573
21	0.9271	42	0.8139	63	0.7253	84	0.6542
22	0.9210	43	0.8092	64	0.7216	85	0.6511
23	0.9150	44	0.8045	65	0.7179	86	0.6481
24	0.9090	45	0.8000	66	0.7142	87	0.6451
25	0.9032	46	0.7954	67	0.7106	88	0.6422
26	0.8974	47	0.7909	68	0.7070	89	0.6392
27	0.8917	48	0.7865	69	0.7035	90	0.6363
28	0.8860	49	0.7821	70	0.7000		
29	0.8805	50	0.7777	71	0.6965		
30	0.8750	51	0.7734	72	0.6930		

FOR LIQUIDS HEAVIER THAN WATER TEMP. 60° FAHR

Beaumé	Specific Gravity.	Beaumé.	Specific Gravity.	Beaumé.	Specific Gravity.	Beaumé.	Specific Gravity.
1	1.0069	19	1.1507	37	1.3425	55	1.6111
2	1.0139	20	1.1600	38	1.3551	56	1.6292
3	1.0211	21	1.1693	39	1.3679	57	1.6477
4	1.0283	22	1.1788	40	1.3809	58	1.6666
5	1.0357	23	1.1885	41	1.3942	59	1.6860
6	1.0431	24	1.1983	42	1.4077	60	1.7056
7	1.0507	25	1.2083	43	1.4215	61	1.7261
8	1.0583	26	1.2184	44	1.4356	62	1.7469
9	1.0661	27	1.2288	45	1.4500	63	1.7682
10	1.0740	28	1.2393	46	1.4646	64	1.7901
11	1.0820	29	1.2500	47	1.4795	65	1.8125
12	1.0902	30	1.2608	48	1.4949	66	1.8354
13	1.0984	31	1.2719	49	1.5104	67	1.8589
14	1.1068	32	1.2831	50	1.5263	68	1.8831
15	1.1153	33	1.2946	51	1.5425	69	1.9079
16	1.1240	34	1.3063	52	1.5591	70	1.9333
17	1.1328	35	1.3181	53	1.5760		
18	1.1417	36	1.3302	54	1.5934		

3. STEAM TABLES.*

(Chap. XI, p. 337.)

TABLE I.—SATURATED STEAM: TEMPERATURE TABLE.

Temp Fahr.	Pressure.		Sp Vol., cu ft per lb.	Density, lbs per cu ft.	Heat of the Liquid	Latent Heat of Evap	Total Heat of Steam	Internal Energy. B.t.u		Entropy.			Temp Fahr.
	Lbs per sq in	Inches of Hg.						Evap.	Steam.	Water	Evap. L/T	Steam	
t	p	—	v or s	$1/v$	h or q	L or r	H	I or ρ	E	n or θ	or r/T	N or ϕ	t
32°	0.0880	0.1804	3204.	0.000301	0.00	1073.4	1073.4	1010.3	1019.3	0.0000	2.1832	2.1832	32°
33	0.0922	0.1878	3170	0.000316	1.01	1072.8	1073.8	1018.6	1019.6	0.0020	2.1777	2.1797	33
34	0.0960	0.1955	3052	0.000328	2.01	1072.2	1074.2	1018.0	1020.0	0.0041	2.1721	2.1762	34
35°	0.0999	0.2034	2938	0.000340	3.02	1071.7	1074.7	1017.3	1020.3	0.0062	2.1660	2.1728	35°
36	0.1040	0.2117	2829	0.000353	4.03	1071.1	1075.1	1016.6	1020.7	0.0082	2.1611	2.1693	36
37	0.1081	0.2202	2725	0.000367	5.04	1070.6	1075.6	1016.0	1021.0	0.0102	2.1557	2.1659	37
38	0.1125	0.2290	2626	0.000381	6.04	1070.0	1076.0	1015.3	1021.3	0.0122	2.1503	2.1625	38
39	0.1170	0.2382	2530	0.000395	7.05	1069.4	1076.5	1014.6	1021.7	0.0142	2.1449	2.1591	39
40°	0.1217	0.2477	2438	0.000410	8.05	1068.9	1076.9	1014.0	1022.0	0.0162	2.1394	2.1556	40°
41	0.1265	0.2575	2350	0.000425	9.05	1068.3	1077.4	1013.3	1022.3	0.0182	2.1341	2.1523	41
42	0.1315	0.2677	2266	0.000441	10.06	1067.8	1077.8	1012.6	1022.7	0.0202	2.1287	2.1489	42
43	0.1366	0.2782	2185	0.000458	11.06	1067.2	1078.3	1012.0	1023.0	0.0222	2.1234	2.1456	43
44	0.1420	0.2890	2107	0.000475	12.06	1066.7	1078.7	1011.3	1023.4	0.0242	2.1181	2.1423	44
45°	0.1475	0.3002	2033	0.000492	13.07	1066.1	1079.2	1010.6	1023.7	0.0262	2.1127	2.1389	45°
46	0.1532	0.3118	1961	0.000510	14.07	1065.6	1079.6	1010.0	1024.0	0.0282	2.1074	2.1356	46
47	0.1591	0.3238	1892	0.000529	15.07	1065.0	1080.1	1009.3	1024.4	0.0301	2.1022	2.1323	47
48	0.1051	0.3363	1820	0.000548	16.07	1064.5	1080.5	1008.6	1024.7	0.0321	2.0970	2.1291	48
49	0.1715	0.3492	1763	0.000567	17.08	1063.9	1081.0	1007.0	1025.0	0.0341	2.0917	2.1258	49
50°	0.1780	0.3625	1702	0.000587	18.08	1063.3	1081.4	1007.3	1025.4	0.0361	2.0865	2.1226	50°
51	0.1848	0.3762	1643	0.000608	19.08	1062.8	1081.9	1006.6	1025.7	0.0381	2.0814	2.1195	51
52	0.1917	0.3903	1586	0.000630	20.08	1062.2	1082.3	1006.0	1026.1	0.0401	2.0763	2.1161	52
53	0.1989	0.4049	1532	0.000653	21.08	1061.7	1082.7	1005.3	1026.4	0.0420	2.0712	2.1132	53
54	0.2063	0.4201	1480	0.000676	22.08	1061.1	1083.2	1004.6	1026.7	0.0440	2.0660	2.1100	54
55°	0.2140	0.4357	1430	0.000700	23.08	1060.6	1083.6	1004.0	1027.1	0.0459	2.0609	2.1068	55°
56	0.2219	0.4518	1381	0.000724	24.08	1060.0	1084.1	1003.3	1027.4	0.0478	2.0559	2.1037	56
57	0.2301	0.4684	1335	0.000749	25.08	1059.5	1084.5	1002.7	1027.7	0.0498	2.0508	2.1006	57
58	0.2385	0.4856	1291	0.000775	26.08	1058.9	1085.0	1002.0	1028.1	0.0517	2.0458	2.0975	58
59	0.2472	0.5034	1249	0.000801	27.08	1058.3	1085.4	1001.3	1028.4	0.0536	2.0408	2.0944	59
60°	0.2562	0.522	1208	0.000828	28.08	1057.8	1085.9	1000.7	1028.7	0.0555	2.0358	2.0913	60°
61	0.2654	0.541	1168	0.000856	29.08	1057.2	1086.3	1000.0	1029.1	0.0574	2.0308	2.0882	61
62	0.2749	0.560	1130	0.000885	30.08	1056.7	1086.8	999.3	1029.4	0.0593	2.0258	2.0851	62
63	0.2847	0.580	1093	0.000915	31.07	1056.1	1087.2	998.7	1029.8	0.0612	2.0209	2.0821	63
64	0.2949	0.601	1058	0.000946	32.07	1055.6	1087.6	998.0	1030.1	0.0631	2.0160	2.0791	64
65°	0.3054	0.622	1024	0.000977	33.07	1055.0	1088.1	997.4	1030.4	0.0650	2.0110	2.0760	65
66	0.3161	0.644	991	0.001009	34.07	1054.5	1088.5	996.7	1030.8	0.0669	2.0062	2.0731	66
67	0.3272	0.667	959	0.001043	35.07	1053.9	1089.0	996.0	1031.1	0.0688	2.0013	2.0701	67
68	0.3386	0.690	928	0.001077	36.07	1053.4	1089.4	995.4	1031.4	0.0707	1.9965	2.0672	68
69	0.3504	0.714	899	0.001112	37.06	1052.8	1089.9	994.7	1031.8	0.0726	1.9916	2.0642	69
70°	0.3626	0.739	871	0.001148	38.06	1052.3	1090.3	994.0	1032.1	0.0745	1.9868	2.0613	70°
71	0.3751	0.764	843	0.001186	39.06	1051.7	1090.8	993.4	1032.4	0.0764	1.9821	2.0585	71
72	0.3880	0.790	817	0.001224	40.05	1051.2	1091.2	992.7	1032.8	0.0783	1.9773	2.0556	72
73	0.4012	0.817	792.	0.001263	41.05	1050.6	1091.6	992.0	1033.1	0.0802	1.9726	2.0528	73
74	0.4148	0.845	767.	0.001304	42.05	1050.0	1092.1	991.4	1033.4	0.0821	1.9678	2.0499	74

$T^\circ = t^\circ + 459.6$; $J = 777.5$ ft.-lbs per B.t.u [log = 2.89071], $A = 1/J = 1.286 \times 10^{-3}$; $144\ A = 0.1852$ [log = 1.26764]

For water, at 45° (0.15 lb), sp. vol., v' or $\sigma = 0.01602$ cu ft per lb; $1/v' = 62.4$ lbs. per cu. ft; $144\ Apv' = 0.0004$ B.t.u

For water, at 70° (0.36 lb.); sp. vol., v' or $\sigma = 0.01605$ cu. ft per lb.; $1/v' = 62.3$ lbs. per cu. ft; $144\ Apv' = 0.001$ B.t.u.

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STEAM TABLES.—*Continued.*

TABLE I.—TEMPERATURE TABLE.

Temp Fahr.	Pressure.		Sp. Vol., cu. ft. per lb.	Density, lbs. per cu. ft.	Heat of the Liquid	Latent Heat of Evap.	Total Heat of Steam.	Internal Energy. B.t.u.		Entropy.			Temp Fahr.
	Lbs. per sq. in.	Inches of Hg.						Evap.	Steam.	Water	Evap. L/T	Steam	
t	p	—	v or s	$1/v$	h or q	L or r	H	I or ρ	E	n or θ	or r/T	N or ϕ	t
75°	0.4288	0.873	743	0.001346	43.05	1040.5	1092.5	990.7	1033.8	0.0840	1.9631	2.0471	75°
76	0.4432	0.903	720	0.001389	44.04	1048.9	1093.0	990.1	1034.1	0.0858	1.9585	2.0443	76
77	0.4581	0.933	698	0.001433	45.04	1048.4	1093.4	989.4	1031.4	0.0876	1.9538	2.0414	77
78	0.4735	0.964	677.	0.001477	46.04	1047.8	1003.9	988.7	1034.8	0.0895	1.9491	2.0386	78
79	0.4893	0.996	657.	0.001523	47.04	1047.3	1094.3	988.1	1035.1	0.0913	1.9445	2.0358	79
80°	0.505	1.029	636.8	0.001570	48.03	1046.7	1094.8	987.4	1035.4	0.0932	1.9398	2.0330	80°
81	0.522	1.063	617.5	0.001619	49.03	1046.2	1095.2	986.7	1035.8	0.0950	1.9352	2.0302	81
82	0.539	1.098	598.7	0.001670	50.03	1045.6	1095.6	986.1	1036.1	0.0969	1.9306	2.0275	82
83	0.557	1.134	580.5	0.001723	51.02	1045.1	1096.1	985.4	1036.4	0.0987	1.9260	2.0247	83
84	0.575	1.171	562.9	0.001777	52.02	1044.5	1096.5	984.8	1036.8	0.1005	1.9215	2.0220	84
85°	0.594	1.209	545.9	0.001832	53.02	1044.0	1097.0	984.1	1037.1	0.1023	1.9169	2.0192	85°
86	0.613	1.248	529.5	0.001889	54.01	1043.4	1007.4	983.4	1037.4	0.1041	1.9124	2.0165	86
87	0.633	1.289	513.7	0.001947	55.01	1042.8	1007.0	982.8	1037.8	0.1060	1.9079	2.0139	87
88	0.654	1.331	498.4	0.002007	56.01	1042.2	1008.3	982.1	1038.1	0.1078	1.9034	2.0112	88
89	0.675	1.373	483.6	0.002068	57.00	1041.7	1098.7	981.4	1038.4	0.1096	1.8989	2.0085	89
90°	0.696	1.417	469.3	0.002131	58.00	1041.2	1090.2	980.8	1038.8	0.1114	1.8944	2.0058	90°
91	0.718	1.462	455.5	0.002195	59.00	1040.6	1090.6	980.1	1039.1	0.1133	1.8900	2.0033	91
92	0.741	1.508	442.2	0.002261	60.00	1010.0	1100.1	979.4	1039.4	0.1151	1.8856	2.0007	92
93	0.765	1.556	429.4	0.002329	60.99	1039.5	1100.5	978.8	1039.8	0.1169	1.8812	1.9981	93
94	0.789	1.605	417.0	0.002398	61.99	1039.0	1101.0	978.1	1040.1	0.1187	1.8767	1.9954	94
95°	0.813	1.655	405.0	0.002469	62.99	1038.4	1101.4	977.4	1040.4	0.1205	1.8723	1.9928	95°
96	0.838	1.706	393.4	0.002542	63.98	1037.8	1101.8	976.8	1040.8	0.1223	1.8680	1.9903	96
97	0.864	1.759	382.2	0.002617	64.98	1037.3	1102.3	976.1	1041.1	0.1241	1.8636	1.9877	97
98	0.891	1.813	371.4	0.002693	65.98	1036.7	1102.8	975.5	1041.4	0.1259	1.8592	1.9851	98
99	0.918	1.869	360.9	0.002771	66.97	1036.2	1103.2	974.8	1041.8	0.1277	1.8549	1.9826	99
100°	0.946	1.926	350.8	0.002851	67.97	1035.6	1103.6	974.1	1042.1	0.1295	1.8505	1.9800	100°
101	0.975	1.985	341.0	0.002933	68.97	1035.1	1104.0	973.5	1012.4	0.1313	1.8463	1.9776	101
102	1.005	2.045	331.5	0.003017	69.96	1034.5	1104.5	972.8	1042.8	0.1330	1.8420	1.9750	102
103	1.035	2.107	322.2	0.003104	70.96	1034.0	1104.9	972.1	1043.1	0.1347	1.8377	1.9724	103
104	1.066	2.171	313.3	0.003192	71.96	1033.4	1105.3	971.5	1013.4	0.1365	1.8335	1.9700	104
105°	1.098	2.236	304.7	0.003282	72.95	1032.8	1105.8	970.8	1043.8	0.1383	1.8292	1.9675	105°
106	1.131	2.303	296.4	0.003374	73.95	1032.3	1106.2	970.1	1044.1	0.1401	1.8250	1.9651	106
107	1.165	2.372	288.3	0.003469	74.95	1031.7	1106.7	969.5	1044.4	0.1418	1.8208	1.9626	107
108	1.199	2.443	280.5	0.003565	75.95	1031.2	1107.1	968.8	1044.8	0.1436	1.8166	1.9602	108
109	1.235	2.515	272.9	0.003664	76.94	1030.6	1107.5	968.2	1045.1	0.1454	1.8124	1.9578	109
110°	1.271	2.589	265.5	0.003766	77.94	1030.0	1108.0	967.5	1045.4	0.1471	1.8082	1.9553	110°
111	1.308	2.665	258.3	0.003871	78.94	1029.5	1108.4	966.8	1045.8	0.1489	1.8041	1.9530	111
112	1.346	2.740	251.4	0.003978	79.93	1028.9	1108.8	966.2	1046.1	0.1506	1.8000	1.9506	112
113	1.386	2.822	244.7	0.004087	80.93	1028.4	1109.3	965.5	1046.4	0.1524	1.7959	1.9483	113
114	1.426	2.904	238.2	0.004198	81.93	1027.8	1109.7	964.8	1046.8	0.1541	1.7917	1.9458	114
115°	1.467	2.987	231.9	0.004312	82.92	1027.2	1110.2	964.2	1047.1	0.1559	1.7876	1.9435	115°
116	1.509	3.073	225.8	0.004429	83.92	1026.7	1110.6	963.5	1047.4	0.1576	1.7836	1.9412	116
117	1.553	3.161	219.9	0.004548	84.92	1026.1	1111.0	962.8	1047.8	0.1591	1.7795	1.9389	117
118	1.597	3.252	214.1	0.004671	85.92	1025.5	1111.5	962.2	1048.1	0.1611	1.7755	1.9366	118
119	1.642	3.344	208.5	0.004796	86.91	1025.0	1111.9	961.5	1048.4	0.1628	1.7715	1.9343	119
120°	1.689	3.438	203.1	0.004924	87.91	1024.4	1112.3	960.8	1048.7	0.1645	1.7674	1.9319	120°
121	1.736	3.535	197.9	0.005054	88.91	1023.9	1112.8	960.2	1049.1	0.1662	1.7634	1.9296	121
122	1.785	3.635	192.8	0.005187	89.91	1023.3	1113.2	959.5	1049.4	0.1679	1.7594	1.0273	122
123	1.835	3.737	187.9	0.005323	90.90	1022.7	1113.6	958.8	1049.7	0.1696	1.7555	1.9251	123
124	1.886	3.841	183.1	0.005462	91.90	1022.2	1114.1	958.2	1050.0	0.1713	1.7515	1.9228	124

$T^\circ = t^\circ + 459.6$; $J = 777.5$ ft.-lbs per B.t u. [log = 2.89071]; $A = 1/J = 1.286 \times 10^{-3}$; $144\,A = 0.1852$ [log = $\bar{1}.26764$].

For water at 95° (0.81 lb.), sp. vol, v' or $\sigma = 0.0161$ cu. ft per lb; $1/v' = 62.0$ lbs. per cu. ft.; $144\,Apv' = 0.002$ B.t.u.

For water, at 120° (1.69 lbs.), sp vol, v' or $\sigma = 0.0162$ cu. ft. per lb.; $1/v' = 61.7$ lbs. per cu. ft, $144\,Apv' = 0.005$ B.t.u.

STEAM TABLES. — *Continued.*

TABLE I. — TEMPERATURE TABLE.

Temp Fahr.	Pressure.		Sp. Vol., cu. ft. per lb.	Density, lbs. per cu. ft.	Heat of the Liquid	Latent Heat of Evap.	Total Heat of Steam.	Internal Energy B.t.u.		Entropy.			Temp Fahr.
	Lbs per sq in	Inches of Hg.						Evap.	Steam	Water	Evap. L/T	Steam	
t	*p*	–	*v* or *s*	1/*v*	*h* or *q*	*L* or *r*	*H*	*I* or *ρ*	*E*	*n* or *θ*	or *r*/*T*	*N* or *φ*	*t*
125°	1.938	3.948	178.4	0.005605	92.90	1021.6	1114.5	957.5	1050.4	0.1730	1.7475	1.9205	125°
126	1.992	4.057	173.9	0.005751	93.90	1021.1	1115.0	956.8	1050.7	0.1747	1.7436	1.9183	126
127	2.047	4.168	169.6	0.005900	94.89	1020.5	1115.4	956.1	1051.0	0.1764	1.7397	1.9161	127
128	2.103	4.282	165.3	0.006052	95.89	1019.9	1115.8	955.5	1051.3	0.1781	1.7358	1.9139	128
129	2.160	4.399	161.1	0.006207	96.89	1019.4	1116.2	954.8	1051.7	0.1790	1.7318	1.9117	129
130°	2.219	4.52	157.1	0.00537	97.89	1018.8	1116.7	954.1	1052.0	0.1816	1.7279	1.9005	130°
131	2.279	4.64	153.2	0.00553	98.89	1018.2	1117.1	953.4	1052.3	0.1833	1.7240	1.9073	131
132	2.340	4.76	140.4	0.00569	99.88	1017.7	1117.5	952.8	1052.7	0.1849	1.7202	1.9051	132
133	2.403	4.89	145.8	0.00585	100.88	1017.1	1118.0	952.1	1053.0	0.1866	1.7164	1.9030	133
134	2.467	5.02	142.2	0.00703	101.88	1016.5	1118.4	951.4	1053.3	0.1883	1.7125	1.9008	134
135°	2.533	5.16	138.7	0.00721	102.88	1016.0	1118.8	950.8	1053.6	0.1900	1.7086	1.8986	135°
136	2.600	5.29	135.4	0.00739	103.88	1015.4	1119.3	950.1	1054.0	0.1917	1.7048	1.8965	136
137	2.669	5.43	132.1	0.00757	104.87	1014.8	1119.7	949.4	1054.3	0.1933	1.7010	1.8943	137
138	2.740	5.58	128.9	0.00776	105.87	1014.3	1120.1	948.8	1054.6	0.1950	1.6972	1.8922	138
139	2.812	5.73	125.8	0.00795	106.87	1013.7	1120.6	948.1	1055.0	0.1967	1.6934	1.8901	139
140°	2.885	5.88	122.8	0.00814	107.87	1013.1	1121.0	947.4	1055.3	0.1984	1.6896	1.8880	140°
141	2.960	6.03	119.9	0.00834	108.87	1012.6	1121.4	946.8	1055.6	0.2000	1.6859	1.8859	141
142	3.037	6.18	117.1	0.00854	109.87	1012.0	1121.8	946.1	1055.9	0.2017	1.6821	1.8838	142
143	3.115	6.34	114.3	0.00875	110.87	1011.4	1122.3	945.4	1056.3	0.2033	1.6774	1.8817	143
144	3.195	6.51	111.6	0.00896	111.87	1010.8	1122.7	944.7	1056.6	0.2050	1.6746	1.8796	144
145°	3.277	6.67	109.0	0.00918	112.86	1010.3	1123.1	944.0	1056.9	0.2067	1.6709	1.8776	145°
146	3.361	6.84	106.5	0.00940	113.86	1009.7	1123.6	943.4	1057.2	0.2083	1.6672	1.8755	146
147	3.446	7.02	104.0	0.00962	114.86	1009.1	1124.0	942.7	1057.6	0.2100	1.6635	1.8735	147
148	3.533	7.20	101.6	0.00985	115.86	1008.6	1124.4	942.0	1057.9	0.2116	1.6598	1.8714	148
149	3.623	7.38	99.2	0.01008	116.86	1008.0	1124.8	941.4	1058.2	0.2132	1.6562	1.8694	149
150°	3.714	7.57	96.9	0.01032	117.86	1007.4	1125.3	940.7	1058.5	0.2149	1.6525	1.8674	150°
151	3.809	7.76	94.7	0.01056	118.86	1006.8	1125.7	940.0	1058.8	0.2165	1.6488	1.8653	151
152	3.902	7.95	92.6	0.01080	119.86	1006.2	1126.1	930.3	1059.2	0.2182	1.6452	1.8634	152
153	3.999	8.14	90.5	0.01105	120.86	1005.7	1126.5	938.6	1050.5	0.2198	1.6416	1.8614	153
154	4.098	8.34	88.4	0.01131	121.86	1005.1	1127.0	938.0	1059.8	0.2214	1.6380	1.8594	154
155°	4.199	8.55	86.4	0.01157	122.86	1004.5	1127.4	937.3	1060.2	0.2231	1.6344	1.8575	155°
156	4.303	8.76	84.5	0.01184	123.86	1003.9	1127.8	936.6	1060.5	0.2247	1.6308	1.8555	156
157	4.408	8.98	82.6	0.01211	124.86	1003.4	1128.2	935.9	1060.8	0.2263	1.6272	1.8535	157
158	4.515	9.20	80.7	0.01239	125.86	1002.8	1128.6	935.3	1061.1	0.2279	1.6236	1.8515	158
159	4.625	9.42	78.9	0.01267	126.86	1002.2	1129.1	934.6	1061.5	0.2205	1.6201	1.8496	159
160°	4.737	9.65	77.2	0.01296	127.86	1001.6	1129.5	933.9	1061.8	0.2311	1.6165	1.8476	160°
161	4.851	9.88	75.5	0.01325	128.86	1001.1	1129.9	933.3	1062.1	0.2327	1.6130	1.8457	161
162	4.967	10.12	73.8	0.01355	129.86	1000.5	1130.4	932.6	1062.4	0.2344	1.6094	1.8438	162
163	5.086	10.36	72.2	0.01386	130.86	999.9	1130.8	931.9	1062.8	0.2360	1.6059	1.8419	163
164	5.208	10.61	70.6	0.01417	131.86	999.3	1131.2	931.2	1063.1	0.2376	1.6024	1.8400	164

$T^\circ = t^\circ + 459.6$; $J = 777.5$ ft.-lbs per B.t.u [log = 2.89071]; $A = 1/J = 1.286 \times 10^{-3}$; 144 A = 0.1852 [log = $\bar{1}.26764$].

For water, at 145° (3.28 lbs.), sp. vol., v' or σ = 0.0163 cu ft. per lb.; $1/v'$ = 61.3 lbs per cu. ft.; 144 Apv' = 0.01 B.t.u.

For water, at 170° (5.99 lbs.), sp. vol., v' or σ = 0.0164 cu. ft. per lb.; $1/v'$ = 60.8 lbs. per cu. ft.; 144 Apv' = 0.02 B.t.u.

STEAM TABLES. — *Continued.*

TABLE II. — SATURATED STEAM: PRESSURE TABLE.

Press. lbs.	Temp. Deg. F.	Press. Atmos.*	Sp. Vol., cu. ft. per lb.	Density, lbs. per cu. ft.	Heat of the Liquid	Latent Heat of Evap.	Total Heat of Steam	Internal Energy. B.t.u.		Entropy.			Press. lbs.
								Evap.	Steam.	Water.	Evap. L/T	Steam.	
p	t	—	v or s	$1/v$	h or q	L or r	H	I or ρ	E	n or θ	or r/T	N or ϕ	p
1	101.83	0.068	333.0	0.00300	69.8	1034.6	1104.4	972.9	1042.7	0.1327	1.8427	1.9754	1
2	126.15	0.136	173.5	0.00576	94.0	1021.0	1115.0	956.7	1050.7	0.1749	1.7431	1.9180	2
3	141.52	0.204	118.5	0.00845	109.4	1012.3	1121.6	946.4	1055.8	0.2008	1.6840	1.8848	3
4	153.01	0.272	90.5	0.01107	120.9	1005.7	1126.5	938.6	1059.5	0.2198	1.6416	1.8614	4
5	162.28	0.340	73.33	0.01364	130.1	1000.3	1130.5	932.4	1062.5	0.2348	1.6084	1.8432	5
6	170.06	0.408	61.89	0.01616	137.9	995.8	1133.7	927.0	1064.9	0.2471	1.5814	1.8285	6
7	176.85	0.476	53.56	0.01867	144.7	991.8	1136.5	922.4	1067.1	0.2579	1.5582	1.8161	7
8	182.86	0.544	47.27	0.02115	150.8	988.2	1139.0	918.2	1069.0	0.2673	1.5380	1.8053	8
9	188.27	0.612	42.36	0.02361	156.2	985.0	1141.1	914.4	1070.5	0.2756	1.5202	1.7958	9
10	193.22	0.680	38.38	0.02606	161.1	982.0	1143.1	910.9	1072.0	0.2832	1.5042	1.7874	10
11	197.75	0.748	35.10	0.02849	165.7	979.2	1144.9	907.8	1073.4	0.2902	1.4895	1.7797	11
12	201.96	0.816	32.36	0.03090	169.9	976.6	1146.5	904.8	1074.7	0.2967	1.4760	1.7727	12
13	205.87	0.885	30.03	0.03330	173.8	974.2	1148.0	902.0	1075.8	0.3025	1.4639	1.7664	13
14	209.55	0.953	28.02	0.03569	177.5	971.9	1149.4	899.3	1076.8	0.3081	1.4523	1.7604	14
15	213.0	1.021	26.27	0.03806	181.0	969.7	1150.7	896.8	1077.8	0.3133	1.4416	1.7549	15
16	216.3	1.089	24.79	0.04042	184.4	967.6	1152.0	894.4	1078.7	0.3183	1.4311	1.7494	16
17	219.4	1.157	23.38	0.04277	187.5	965.6	1153.1	892.1	1079.6	0.3229	1.4215	1.7444	17
18	222.4	1.225	22.16	0.04512	190.5	963.7	1154.2	889.9	1080.4	0.3273	1.4127	1.7400	18
19	225.2	1.293	21.07	0.04746	193.4	961.8	1155.2	887.8	1081.1	0.3315	1.4045	1.7360	19
20	228.0	1.361	20.08	0.04980	196.1	960.0	1156.2	885.8	1081.9	0.3355	1.3965	1.7320	20
21	230.6	1.429	19.18	0.05213	198.8	958.3	1157.1	883.9	1082.6	0.3393	1.3887	1.7280	21
22	233.1	1.497	18.37	0.05445	201.3	956.7	1158.0	882.0	1083.2	0.3430	1.3811	1.7241	22
23	235.5	1.565	17.62	0.05676	203.8	955.1	1158.8	880.2	1083.9	0.3465	1.3739	1.7204	23
24	237.8	1.633	16.93	0.05907	206.1	953.5	1159.6	878.5	1084.5	0.3499	1.3670	1.7169	24
25	240.1	1.701	16.30	0.0614	208.4	952.0	1160.4	876.8	1085.1	0.3532	1.3604	1.7136	25
26	242.2	1.769	15.72	0.0636	210.6	950.6	1161.2	875.1	1085.6	0.3564	1.3542	1.7106	26
27	244.4	1.837	15.18	0.0659	212.7	949.2	1161.9	873.5	1086.2	0.3594	1.3483	1.7077	27
28	246.4	1.905	14.67	0.0682	214.8	947.8	1162.6	872.0	1086.7	0.3623	1.3425	1.7048	28
29	248.4	1.973	14.19	0.0705	216.8	946.4	1163.2	870.5	1087.2	0.3652	1.3367	1.7019	29
30	250.3	2.041	13.74	0.0728	218.8	945.1	1163.9	869.0	1087.7	0.3680	1.3311	1.6991	30
31	252.2	2.109	13.32	0.0751	220.7	943.8	1164.5	867.6	1088.2	0.3707	1.3257	1.6964	31
32	254.1	2.178	12.93	0.0773	222.6	942.5	1165.1	866.2	1088.6	0.3733	1.3205	1.6938	32
33	255.8	2.246	12.57	0.0795	224.4	941.3	1165.7	864.8	1089.1	0.3759	1.3155	1.6914	33
34	257.6	2.314	12.22	0.0818	226.2	940.1	1166.3	863.4	1089.5	0.3784	1.3107	1.6891	34
35	259.3	2.382	11.89	0.0841	227.9	938.9	1166.8	862.1	1089.9	0.3808	1.3060	1.6868	35
36	261.0	2.450	11.58	0.0863	229.6	937.7	1167.3	860.8	1090.3	0.3832	1.3014	1.6846	36
37	262.6	2.518	11.29	0.0886	231.3	936.6	1167.8	859.5	1090.7	0.3855	1.2969	1.6824	37
38	264.2	2.586	11.01	0.0908	232.9	935.5	1168.4	858.3	1091.0	0.3877	1.2925	1.6802	38
39	265.8	2.654	10.74	0.0931	234.5	934.4	1168.9	857.1	1091.4	0.3899	1.2882	1.6781	39
40	267.3	2.722	10.49	0.0953	236.1	933.3	1169.4	855.9	1091.8	0.3920	1.2841	1.6761	40
41	268.7	2.790	10.25	0.0976	237.6	932.2	1169.8	854.7	1092.2	0.3941	1.2800	1.6741	41
42	270.2	2.858	10.02	0.0998	239.1	931.2	1170.3	853.6	1092.5	0.3962	1.2759	1.6721	42
43	271.7	2.926	9.80	0.1020	240.5	930.2	1170.7	852.4	1092.8	0.3982	1.2720	1.6702	43
44	273.1	2.994	9.59	0.1043	242.0	929.2	1171.2	851.3	1093.2	0.4002	1.2681	1.6683	44
45	274.5	3.062	9.39	0.1065	243.4	928.2	1171.6	850.3	1093.5	0.4021	1.2644	1.6665	45
46	275.8	3.130	9.20	0.1087	244.8	927.2	1172.0	849.2	1093.8	0.4040	1.2607	1.6647	46
47	277.2	3.198	9.02	0.1109	246.1	926.3	1172.4	848.1	1094.1	0.4059	1.2571	1.6630	47
48	278.5	3.266	8.84	0.1131	247.5	925.3	1172.8	847.1	1094.4	0.4077	1.2536	1.6613	48
49	279.8	3.334	8.67	0.1153	248.8	924.4	1173.2	846.1	1094.7	0.4095	1.2502	1.6597	49

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* 1 atmo (standard atmosphere) = 760 mms. of Hg. by def. = 29.921 ins. of Hg. = 14.696 lbs. per sq. in.

$T° = t° + 459.6$; $J = 777.5$ ft.-lbs. per B.t.u. [log = 2.89071]; $A = 1/J = 1.286 \times 10^{-3}$; $144\,A = 0.1852$ [log = $\bar{1}.26764$]

For water, at 15 lbs., sp. vol., v' or σ = 0.0167 cu. ft. per lb.; $1/v'$ = 59.8 lbs. per cu. ft.; $144\,Apv'$ = 0.05 B.t.u.

For water, at 40 lbs., sp. vol., v' or σ = 0.0171 cu. ft. per lb.; $1/v'$ = 58.3 lbs. per cu. ft.; $144\,Apv'$ = 0.13 B.t.u.

STEAM TABLES. — *Continued.*

TABLE II. — PRESSURE TABLE.

Press. lbs.	Temp Deg. F.	Press Atmos.*	Sp Vol., cu ft. per lb.	Density, lbs per cu. ft.	Heat of the Liquid	Latent Heat of Evap	Total Heat of Steam.	Internal Energy. B.t u.		Entropy.			Press. lbs.
								Evap.	Steam	Water	Evap L/T	Steam.	
p	*t*	—	*v* or *s*	$1/v$	*h* or *q*	*L* or *r*	*H*	*I* or *ρ*	*E*	*n* or *θ*	or r/T	*N* or *φ*	*p*
50	281.0	3.402	8.51	0.1175	250.1	923.5	1173.6	845.0	1095.0	0.4113	1.2468	1.6581	**50**
51	282.3	3.470	8.35	0.1197	251.4	922.6	1174.0	844.0	1095.3	0.4130	1.2435	1.6565	**51**
52	283.5	3.538	8.20	0.1219	252.6	921.7	1174.3	843.1	1095.5	0.4147	1.2402	1.6549	**52**
53	284.7	3.606	8.05	0.1241	253.9	920.8	1174.7	842.1	1095.8	0.4164	1.2370	1.6534	**53**
54	285.9	3.674	7.91	0.1263	255.1	919.9	1175.0	841.1	1096.1	0.4180	1.2339	1.6519	**54**
55	287.1	3.742	7.78	0.1285	256.3	919.0	1175.4	840.2	1096.3	0.4196	1.2309	1.6505	**55**
56	288.2	3.810	7.65	0.1307	257.5	918.2	1175.7	839.3	1096.6	0.4212	1.2278	1.6490	**56**
57	289.4	3.878	7.52	0.1329	258.7	917.4	1176.0	838.3	1096.8	0.4227	1.2248	1.6475	**57**
58	290.5	3.947	7.40	0.1350	259.8	916.5	1176.4	837.4	1097.1	0.4242	1.2218	1.6460	**58**
59	291.6	4.015	7.28	0.1372	261.0	915.7	1176.7	836.5	1097.3	0.4257	1.2189	1.6446	**59**
60	292.7	4.083	7.17	0.1394	262.1	914.9	1177.0	835.6	1097.6	0.4272	1.2160	1.6432	**60**
61	293.8	4.151	7.06	0.1416	263.2	914.1	1177.3	834.8	1097.8	0.4287	1.2132	1.6419	**61**
62	294.9	4.219	6.95	0.1438	264.3	913.3	1177.6	833.9	1098.0	0.4302	1.2104	1.6406	**62**
63	295.9	4.287	6.85	0.1460	265.4	912.5	1177.9	833.1	1098.2	0.4316	1.2077	1.6393	**63**
64	297.0	4.355	6.75	0.1482	266.4	911.8	1178.2	832.2	1098.4	0.4330	1.2050	1.6380	**64**
65	298.0	4.423	6.65	0.1503	267.5	911.0	1178.5	831.4	1098.7	0.4344	1.2034	1.6368	**65**
66	299.0	4.491	6.56	0.1525	268.5	910.2	1178.8	830.5	1098.9	0.4358	1.2007	1.6355	**66**
67	300.0	4.559	6.47	0.1547	269.6	909.5	1179.0	829.7	1099.1	0.4371	1.1972	1.6343	**67**
68	301.0	4.627	6.38	0.1569	270.6	908.7	1179.3	828.9	1099.3	0.4385	1.1946	1.6331	**68**
69	302.0	4.695	6.29	0.1590	271.6	908.0	1179.6	828.1	1099.5	0.4398	1.1921	1.6319	**69**
70	302.9	4.763	6.20	0.1612	272.6	907.2	1179.8	827.3	1099.7	0.4411	1.1896	1.6307	**70**
71	303.9	4.831	6.12	0.1634	273.6	906.5	1180.1	826.5	1099.9	0.4424	1.1872	1.6296	**71**
72	304.8	4.899	6.04	0.1656	274.5	905.8	1180.4	825.8	1100.1	0.4437	1.1848	1.6285	**72**
73	305.8	4.967	5.96	0.1678	275.5	905.1	1180.6	825.0	1100.3	0.4449	1.1825	1.6274	**73**
74	306.7	5.035	5.89	0.1699	276.5	904.4	1180.9	824.2	1100.5	0.4462	1.1801	1.6263	**74**
75	307.6	5.103	5.81	0.1721	277.4	903.7	1181.1	823.5	1100.6	0.4474	1.1778	1.6252	**75**
76	308.5	5.171	5.74	0.1743	278.3	903.0	1181.4	822.7	1100.8	0.4487	1.1755	1.6242	**76**
77	309.4	5.239	5.67	0.1764	279.3	902.3	1181.6	822.0	1101.0	0.4499	1.1732	1.6231	**77**
78	310.3	5.307	5.60	0.1786	280.2	901.7	1181.8	821.3	1101.2	0.4511	1.1710	1.6221	**78**
79	311.2	5.375	5.54	0.1808	281.1	901.0	1182.1	820.6	1101.4	0.4523	1.1687	1.6210	**79**
80	312.0	5.444	5.47	0.1829	282.0	900.3	1182.3	819.8	1101.6	0.4535	1.1665	1.6200	**80**
81	312.9	5.512	5.41	0.1851	282.9	899.7	1182.5	819.1	1101.7	0.4546	1.1644	1.6190	**81**
82	313.8	5.580	5.34	0.1873	283.8	899.0	1182.8	818.4	1101.9	0.4557	1.1623	1.6180	**82**
83	314.6	5.648	5.28	0.1894	284.6	898.4	1183.0	817.7	1102.1	0.4568	1.1602	1.6170	**83**
84	315.4	5.716	5.22	0.1915	285.5	897.7	1183.2	817.0	1102.2	0.4579	1.1581	1.6160	**84**
85	316.3	5.784	5.16	0.1937	286.3	897.1	1183.4	816.3	1102.4	0.4590	1.1561	1.6151	**85**
86	317.1	5.852	5.10	0.1959	287.2	896.4	1183.6	815.6	1102.6	0.4601	1.1540	1.6141	**86**
87	317.9	5.920	5.05	0.1980	288.0	895.8	1183.8	815.0	1102.7	0.4612	1.1520	1.6132	**87**
88	318.7	5.988	5.00	0.2001	288.9	895.2	1184.0	814.3	1102.9	0.4623	1.1500	1.6123	**88**
89	319.5	6.056	4.94	0.2023	289.7	894.6	1184.2	813.6	1103.0	0.4633	1.1481	1.6114	**89**
90	320.3	6.124	4.89	0.2044	290.5	893.9	1184.4	813.0	1103.2	0.4644	1.1461	1.6105	**90**
91	321.1	6.192	4.84	0.2065	291.3	893.3	1184.6	812.3	1103.3	0.4654	1.1442	1.6096	**91**
92	321.8	6.260	4.79	0.2087	292.1	892.7	1184.8	811.7	1103.5	0.4664	1.1423	1.6087	**92**
93	322.6	6.328	4.74	0.2109	292.9	892.1	1185.0	811.0	1103.6	0.4674	1.1404	1.6078	**93**
94	323.4	6.396	4.69	0.2130	293.7	891.5	1185.2	810.4	1103.8	0.4684	1.1385	1.6069	**94**
95	324.1	6.464	4.65	0.2151	294.5	890.9	1185.4	809.7	1103.9	0.4694	1.1367	1.6061	**95**
96	324.9	6.532	4.60	0.2172	295.3	890.3	1185.6	809.1	1104.1	0.4704	1.1348	1.6052	**96**
97	325.6	6.600	4.56	0.2193	296.1	889.7	1185.8	808.5	1104.2	0.4714	1.1330	1.6044	**97**
98	326.4	6.668	4.51	0.2215	296.8	889.2	1186.0	807.9	1104.4	0.4724	1.1312	1.6036	**98**
99	327.1	6.736	4.47	0.2237	297.6	888.6	1186.2	807.2	1104.5	0.4733	1.1295	1.6028	**99**

* 1 atmo. (standard atmosphere) = 760 mms of Hg. by def. = 29.921 ins. of Hg. = 14.696 lbs per sq. in.

$T^\circ = t^\circ + 459.6$, $J = 777.5$ ft-lbs per B.t.u. [log = 2.89071]; $A = 1/J = 1.286 \times 10^{-3}$; 144 A = 0.1852 [log = $\bar{1}.26764$]

For water, at 65 lbs., sp. vol., v' or σ = 0.0174 cu ft. per lb; $1/v'$ = 57.4 lbs per cu ft.; 144 Apv' = 0.21 B.t.u.

For water, at 90 lbs., sp. vol., v' or σ = 0.0175 cu. ft. per lb.; $1/v'$ = 56.8 lbs. per cu ft.; 144 Apv' = 0.30 B.t.u.

STEAM TABLES. — *Continued.*

TABLE II. — PRESSURE TABLE.

Press lbs.	Temp Deg. F.	Press. Atmos.*	Sp. Vol., cu ft per lb.	Density, lbs. per cu ft.	Heat of the Liquid	Latent Heat of Evap.	Total Heat of Steam	Internal Energy. B.t.u		Entropy.			Press. lbs.
								Evap.	Steam.	Water	Evap L/T	Steam.	
p	t	—	v or s	$1/v$	h or q	L or r	H	I or ρ	E	n or θ	or r/T	N or ϕ	p
100	327.8	6.80	4.429	0.2258	298.3	888.0	1186.3	806.6	1104.6	0.4743	1.1277	1.6020	**100**
101	328.6	6.87	4.388	0.2279	299.1	887.4	1186.5	806.0	1104.8	0.4752	1.1260	1.6012	**101**
102	329.3	6.94	4.347	0.2300	299.8	886.9	1186.7	805.4	1104.9	0.4762	1.1242	1.6004	**102**
103	330.0	7.01	4.307	0.2322	300.6	886.3	1186.9	804.8	1105.0	0.4771	1.1225	1.5996	**103**
104	330.7	7.08	4.268	0.2343	301.3	885.8	1187.0	804.2	1105.1	0.4780	1.1208	1.5988	**104**
105	331.4	7.14	4.230	0.2365	302.0	885.2	1187.2	803.6	1105.3	0.4789	1.1191	1.5980	**105**
106	332.0	7.21	4.192	0.2386	302.7	884.7	1187.4	803.0	1105.4	0.4798	1.1174	1.5972	**106**
107	332.7	7.28	4.155	0.2408	303.4	884.1	1187.5	802.5	1105.5	0.4807	1.1158	1.5965	**107**
108	333.4	7.35	4.118	0.2420	304.1	883.6	1187.7	801.9	1105.7	0.4816	1.1141	1.5957	**108**
109	334.1	7.42	4.082	0.2450	304.8	883.0	1187.9	801.3	1105.8	0.4825	1.1125	1.5950	**109**
110	334.8	7.49	4.047	0.2472	305.5	882.5	1188.0	800.7	1105.9	0.4834	1.1108	1.5942	**110**
111	335.4	7.55	4.012	0.2493	306.2	881.9	1188.2	800.2	1106.0	0.4843	1.1092	1.5935	**111**
112	336.1	7.62	3.978	0.2514	306.9	881.4	1188.4	799.6	1106.2	0.4852	1.1076	1.5928	**112**
113	336.8	7.69	3.915	0.2535	307.6	880.9	1188.5	799.0	1106.3	0.4860	1.1061	1.5921	**113**
114	337.4	7.76	3.912	0.2556	308.3	880.4	1188.7	798.5	1106.4	0.4869	1.1045	1.5914	**114**
115	338.1	7.83	3.880	0.2577	309.0	879.8	1188.8	797.9	1106.5	0.4877	1.1030	1.5907	**115**
116	338.7	7.89	3.848	0.2599	309.6	879.3	1189.0	797.4	1106.6	0.4886	1.1014	1.5900	**116**
117	339.4	7.96	3.817	0.2620	310.3	878.8	1189.1	796.8	1106.8	0.4894	1.0999	1.5893	**117**
118	340.0	8.03	3.786	0.2641	311.0	878.3	1189.3	796.3	1106.9	0.4903	1.0984	1.5887	**118**
119	340.6	8.10	3.756	0.2662	311.6	877.8	1189.4	795.7	1107.0	0.4911	1.0969	1.5880	**119**
120	341.3	8.17	3.726	0.2683	312.3	877.2	1189.6	795.2	1107.1	0.4919	1.0954	1.5873	**120**
121	341.9	8.23	3.697	0.2705	313.0	876.7	1189.7	794.7	1107.2	0.4927	1.0939	1.5866	**121**
122	342.5	8.30	3.668	0.2726	313.6	876.2	1189.8	794.2	1107.3	0.4935	1.0924	1.5859	**122**
123	343.2	8.37	3.639	0.2748	314.3	875.7	1190.0	793.6	1107.4	0.4943	1.0910	1.5853	**123**
124	343.8	8.44	3.611	0.2769	314.9	875.2	1190.1	793.1	1107.6	0.4951	1.0895	1.5846	**124**
125	344.4	8.50	3.583	0.2791	315.5	874.7	1190.3	792.6	1107.7	0.4959	1.0880	1.5839	**125**
126	345.0	8.57	3.556	0.2812	316.2	874.2	1190.4	792.0	1107.8	0.4967	1.0865	1.5832	**126**
127	345.6	8.64	3.530	0.2833	316.8	873.8	1190.5	791.5	1107.9	0.4974	1.0851	1.5825	**127**
128	346.2	8.71	3.504	0.2854	317.4	873.3	1190.7	791.0	1108.0	0.4982	1.0837	1.5819	**128**
129	346.8	8.78	3.478	0.2875	318.0	872.8	1190.8	790.5	1108.1	0.4990	1.0823	1.5813	**129**
130	347.4	8.85	3.452	0.2897	318.6	872.3	1191.0	790.0	1108.2	0.4998	1.0809	1.5807	**130**
131	348.0	8.91	3.427	0.2918	319.3	871.8	1191.1	789.5	1108.3	0.5005	1.0796	1.5801	**131**
132	348.5	8.98	3.402	0.2939	319.9	871.3	1191.2	789.0	1108.4	0.5013	1.0782	1.5795	**132**
133	349.1	9.05	3.378	0.2960	320.5	870.9	1191.3	788.5	1108.5	0.5020	1.0769	1.5789	**133**
134	349.7	9.12	3.354	0.2981	321.1	870.4	1191.5	788.0	1108.6	0.5028	1.0755	1.5783	**134**
135	350.3	9.19	3.331	0.3002	321.7	869.9	1191.6	787.5	1108.7	0.5035	1.0742	1.5777	**135**
136	350.8	9.25	3.308	0.3023	322.3	869.4	1191.7	787.0	1108.8	0.5043	1.0728	1.5771	**136**
137	351.4	9.32	3.285	0.3044	322.8	869.0	1191.8	786.5	1108.9	0.5050	1.0715	1.5765	**137**
138	352.0	9.39	3.263	0.3065	323.4	868.5	1192.0	786.0	1109.0	0.5057	1.0702	1.5759	**138**
139	352.5	9.46	3.241	0.3086	324.0	868.1	1192.1	785.5	1109.1	0.5064	1.0689	1.5753	**139**
140	353.1	9.53	3.219	0.3107	324.6	867.6	1192.2	785.0	1109.2	0.5072	1.0675	1.5747	**140**
141	353.6	9.59	3.197	0.3129	325.2	867.2	1192.3	784.6	1109.3	0.5079	1.0662	1.5741	**141**
142	354.2	9.66	3.175	0.3150	325.8	866.7	1192.5	784.1	1109.4	0.5086	1.0649	1.5735	**142**
143	354.7	9.73	3.154	0.3171	326.3	866.3	1192.6	783.6	1109.5	0.5093	1.0637	1.5730	**143**
144	355.3	9.80	3.133	0.3192	326.9	865.8	1192.7	783.2	1109.6	0.5100	1.0624	1.5724	**144**
145	355.8	9.87	3.112	0.3213	327.4	865.4	1192.8	782.7	1109.6	0.5107	1.0612	1.5719	**145**
146	356.3	9.93	3.092	0.3234	328.0	864.9	1192.9	782.2	1109.7	0.5114	1.0599	1.5713	**146**
147	356.9	10.00	3.072	0.3255	328.6	864.5	1193.0	781.7	1109.8	0.5121	1.0587	1.5708	**147**
148	357.4	10.07	3.052	0.3276	329.1	864.0	1193.2	781.3	1109.9	0.5128	1.0574	1.5702	**148**
149	357.9	10.14	3.033	0.3297	329.7	863.6	1193.3	780.8	1110.0	0.5135	1.0562	1.5697	**149**

* 1 atmo (standard atmosphere) = 760 mms. of Hg by def. = 29.921 ins. of Hg. = 14.696 lbs per sq. in.

$T° = t° + 450.6$; J = 777.5 ft.-lbs. per B.t.u [log = 2.89071]; $A = 1/J = 1.286 \times 10^{-3}$, $144 A$ = 0.1852 [log = $\bar{1}.26764$]

For water, at 115 lbs., sp vol., v' or σ = 0.0178 cu ft. per lb.; $1/v'$ = 56.0 lbs. per cu. ft; $144 Apv'$ = 0.38 B.t.u

For water, at 140 lbs, sp. vol., v' or σ = 0.0180 cu. ft. per lb.; $1/v'$ = 55.4 lbs. per cu. ft.; $144 Apv'$ = 0.47 B.t.u.

STEAM TABLES. — *Concluded.*

TABLE II. — PRESSURE TABLE.

Press. lbs.	Temp. Deg. F.	Press. Atmos.*	Sp Vol., cu. ft per lb.	Density, lbs. per cu. ft.	Heat of the Liquid	Latent Heat of Evap.	Total Heat of Steam.	Internal Energy., B.t.u.		Entropy.			Press. lbs.
								Evap.	Steam.	Water	Evap. L/T	Steam.	
p	t	—	v or s	$1/v$	h or q	L or r	H	I or ρ	E	n or θ	or r/T	N or ϕ	p
150	358.5	10.21	3.012	0.3320	330.2	863.2	1193.4	780.4	1110.1	0.5142	1.0550	1.5692	150
152	359.5	10.34	2.974	0.3362	331.4	862.3	1193.6	779.4	1110.3	0.5155	1.0525	1.5680	152
154	360.5	10.48	2.938	0.3404	332.4	861.4	1193.8	778.5	1110.4	0.5169	1.0501	1.5670	154
156	361.6	10.61	2.902	0.3446	333.5	860.6	1194.1	777.6	1110.6	0.5182	1.0477	1.5659	156
158	362.6	10.75	2.868	0.3488	334.6	859.7	1194.3	776.7	1110.8	0.5195	1.0454	1.5649	158
160	363.6	10.89	2.834	0.3529	335.6	858.8	1194.5	775.8	1110.9	0.5208	1.0431	1.5639	160
162	364.6	11.02	2.801	0.3570	336.7	858.0	1194.7	775.0	1111.1	0.5220	1.0409	1.5629	162
164	365.6	11.16	2.769	0.3612	337.7	857.2	1194.9	774.1	1111.2	0.5233	1.0387	1.5620	164
166	366.5	11.30	2.737	0.3654	338.7	856.4	1195.1	773.2	1111.4	0.5245	1.0365	1.5610	166
168	367.5	11.43	2.706	0.3696	339.7	855.5	1195.3	772.4	1111.5	0.5257	1.0343	1.5600	168
170	368.5	11.57	2.675	0.3738	340.7	854.7	1195.4	771.5	1111.7	0.5269	1.0321	1.5590	170
172	369.4	11.70	2.645	0.3780	341.7	853.9	1195.6	770.7	1111.8	0.5281	1.0300	1.5581	172
174	370.4	11.84	2.616	0.3822	342.7	853.1	1195.8	769.8	1112.0	0.5293	1.0278	1.5571	174
176	371.3	11.97	2.588	0.3864	343.7	852.3	1196.0	769.0	1112.1	0.5305	1.0257	1.5562	176
178	372.2	12.11	2.560	0.3906	344.7	851.5	1196.2	768.2	1112.3	0.5317	1.0235	1.5552	178
180	373.1	12.25	2.533	0.3948	345.6	850.8	1196.4	767.4	1112.4	0.5328	1.0215	1.5543	180
182	374.0	12.38	2.507	0.3989	346.6	850.0	1196.6	766.6	1112.6	0.5339	1.0195	1.5534	182
184	374.9	12.51	2.481	0.4031	347.6	849.2	1196.8	765.8	1112.7	0.5351	1.0174	1.5525	184
186	375.8	12.66	2.455	0.4073	348.5	848.4	1196.9	765.0	1112.8	0.5362	1.0154	1.5516	186
188	376.7	12.79	2.430	0.4115	349.4	847.7	1197.1	764.2	1113.0	0.5373	1.0134	1.5507	188
190	377.6	12.93	2.406	0.4157	350.4	846.9	1197.3	763.4	1113.1	0.5384	1.0114	1.5498	190
192	378.5	13.06	2.381	0.4199	351.3	846.1	1197.4	762.6	1113.2	0.5395	1.0095	1.5490	192
194	379.3	13.20	2.358	0.4241	352.2	845.4	1197.6	761.8	1113.4	0.5405	1.0076	1.5481	194
196	380.2	13.34	2.335	0.4283	353.1	844.7	1197.8	761.1	1113.5	0.5416	1.0056	1.5472	196
198	381.0	13.47	2.312	0.4325	354.0	843.9	1197.9	760.3	1113.6	0.5426	1.0038	1.5464	198
200	381.9	13.61	2.290	0.437	354.9	843.2	1198.1	759.5	1113.7	0.5437	1.0019	1.5456	200
205	384.0	13.95	2.237	0.447	357.1	841.4	1198.5	757.6	1114.0	0.5463	0.9973	1.5436	205
210	386.0	14.29	2.187	0.457	359.2	839.6	1198.8	755.8	1114.4	0.5488	0.9928	1.5416	210
215	388.0	14.63	2.138	0.468	361.4	837.9	1199.2	754.0	1114.6	0.5513	0.9885	1.5398	215
220	389.9	14.97	2.091	0.478	363.4	836.2	1199.6	752.3	1114.9	0.5538	0.9841	1.5379	220
225	391.9	15.31	2.046	0.489	365.5	834.4	1199.9	750.5	1115.2	0.5562	0.9799	1.5361	225
230	393.8	15.65	2.004	0.499	367.5	832.8	1200.2	748.8	1115.4	0.5586	0.9758	1.5344	230
235	395.6	15.99	1.964	0.509	369.4	831.1	1200.6	747.0	1115.7	0.5610	0.9717	1.5327	235
240	397.4	16.33	1.924	0.520	371.4	829.5	1200.9	745.4	1115.9	0.5633	0.9676	1.5309	240
245	399.3	16.67	1.887	0.530	373.3	827.9	1201.2	743.7	1116.2	0.5655	0.9638	1.5293	245
250	401.1	17.01	1.850	0.541	375.2	826.3	1201.5	742.0	1116.4	0.5676	0.9600	1.5276	250
260	404.5	17.69	1.782	0.561	378.9	823.1	1202.1	738.9	1116.9	0.5719	0.9525	1.5244	260
270	407.9	18.37	1.718	0.582	382.5	820.1	1202.6	735.8	1117.3	0.5760	0.9454	1.5214	270
280	411.2	19.05	1.658	0.603	386.0	817.1	1203.1	732.7	1117.7	0.5800	0.9385	1.5185	280
290	414.4	19.73	1.602	0.624	389.4	814.2	1203.6	729.7	1118.1	0.5840	0.9316	1.5156	290
300	417.5	20.41	1.551	0.645	392.7	811.3	1204.1	726.8	1118.5	0.5878	0.9251	1.5129	300
350	431.9	23.82	1.334	0.750	408.2	797.8	1206.1	713.3	1120.2	0.6053	0.8949	1.5002	350
400	444.8	27.22	1.17	0.86	422	786.	1208.	701	1122	0.621	0.868	1.489	400
450	456.5	30.62	1.04	0.96	435	774	1209	690	1123.	0.635	0.844	1.479	450
500	467.3	34.02	0.93	1.08	448	762	1210	678	1124	0.648	0.822	1.470	500

* 1 atmo. (standard atmosphere) = 760 mms of Hg. by def = 29.921 ins. of Hg. = 14.696 lbs. per sq. in.

$T° = t° + 459.6$; $J = 777.5$ ft.-lbs per B.t.u [log = 2.89071]; $A = 1/J = 1.286 \times 10^{-3}$; $144\,A = 0.1852$ [log = $\bar{1}.26764$].

For water, at 215 lbs., sp. vol., v' or σ = 0.0185 cu. ft. per lb.; $1/v'$ = 54.0 lbs. per cu. ft.; $144\,Apv'$ = 0.74 B.t.u.

For water, at 240 lbs., sp. vol., v' or σ = 0.0186 cu. ft. per lb.; $1/v'$ = 53.6 lbs. per cu. ft.; $144\,Apv'$ = 0.83 B.t.u.

TABLE 4.—HORSE POWER PER POUND MEAN PRESSURE.

(Chap. XVI, p. 647.)

Diameter of Cylinder. Inches.	Speed of Piston in Feet per Minute.										
	100	240	300	350	400	450	500	550	600	650	750
4	038	091	.114	.133	.152	.171	.19	.209	.228	247	.285
4½	048	115	144	.168	192	.216	.24	264	288	.312	360
5	06	.144	.18	.210	240	.270	30	.33	.36	.39	.450
5½	072	.173	216	.252	288	.324	36	.396	.432	.468	540
6	.086	205	256	299	.342	385	.428	471	.513	.555	.641
6½	.102	.245	307	.391	409	.464	.512	.563	614	698	800
7	116	.279	.348	408	466	.524	.583	641	.699	.756	874
7½	134	321	.401	468	.534	.602	669	735	.802	869	1.002
8	152	.365	.456	532	.608	.685	761	837	912	.989	1.121
8½	.172	.413	.516	.602	688	774	.86	.946	1 032	1 118	1.290
9	192	462	577	674	.770	866	.963	1 059	1.154	1 251	1.444
9½	215	.515	644	.751	.859	966	1 074	1 181	1 288	1 395	1 610
10	238	.571	714	833	.952	1 071	1 190	1 309	1.428	1 547	1.785
10½	.262	63	787	919	1 050	1 181	1 313	1 444	1 575	1.706	1 969
11	288	691	864	1 008	1 152	1 296	1.44	1 584	1 728	1 872	2 160
11½	314	.754	.943	1.100	1.257	1 414	1 572	1.720	1 886	2 043	2 357
12	342	820	1 025	1 195	1 366	1 540	1.708	1 880	2 050	2 222	2 564
13	402	964	1 206	1 407	1.608	1 809	2 01	2 211	2 412	2 613	3 015
14	466	1 119	1.398	1 631	1 864	2 097	2 331	2 564	2 797	3 029	3.495
15	535	1 285	1 606	1 873	2 131	2 409	2 677	2 945	3 212	3.479	4 004
16	609	1 461	1 827	2 131	2 436	2 741	3 045	3 349	3 654	3 958	4 567
17	685	1 643	2.054	2.396	2 739	3 081	3 424	3.766	4 108	4 450	5 135
18	771	1 849	2 312	2 697	3 083	3 468	3.854	4.239	4 624	5.009	5 780
19	859	2.061	2 577	3 006	3 436	3 865	4.295	4 724	5 154	5 583	6 442
20	.952	2 202	2 855	3 331	3.807	4 285	4 750	5 234	5.731	6 186	7 138
21	1.049	2 518	3 148	3 672	4 197	4 722	5 247	5 771	6 296	6 820	7 869
22	1 152	2 764	3 455	4 031	4 607	5 183	5 750	6 334	6 911	7.486	8 648
23	1 259	3 021	3 776	4 405	5 035	5.664	6.294	6.923	7 552	8 181	9 44
24	1 370	3 289	4 111	4 797	5.482	6 167	6.853	7.538	8 223	8 908	10 279
25	1 487	3 569	4.461	5 105	5 948	6 692	7 436	8 179	8 923	9 566	11 053
26	1 609	3.861	4 826	5 630	6 435	7 239	8 044	8 848	9 652	10 456	12 065
27	1 733	4 159	5 199	6 066	6 932	7 799	8 666	9 532	10 399	11 265	12 998
28	1 865	4 477	5.596	6 529	7 462	8 395	9 328	10 261	11 193	12 125	13 991
29	2 002	4 805	6 006	7 007	8 008	9 009	10 01	11 011	12 012	13 013	15.015
30	2.142	5.141	6 426	7 497	8 568	9 639	10 71	11 781	12 852	13 923	16 065
31	2 288	5 486	6 865	8 001	9 144	10 287	11.43	12 573	13 716	14 860	17.145
32	2 436	5.846	7 308	8 526	9 744	10 962	12 18	13 398	14 616	15.834	18 270
33	2 590	6 216	7 770	9 065	10 360	11 655	12 959	14 245	15 54	16 835	19 425
34	2 746	6 59	8 238	9 611	10 984	12 357	13 73	15.103	16.476	17 849	20 595
35	2.914	6 993	8.742	10 199	11.656	13.113	14 57	16 027	17.484	18 941	21 855
36	3.084	7 401	9 252	10 794	12 336	13.878	15.42	16.962	18 504	20 046	23 130
37	3.253	7.819	9.774	11.403	13.032	14 861	16 29	17 079	19 548	21 177	24 435
38	3.436	8 246	10 308	12 026	13 744	15.462	17 18	18 898	20 616	22 334	25.770
39	3.620	8 648	10 86	12 670	14 480	16 290	18 10	19.91	21 62	23 53	27 150
40	3 808	9 139	11 424	13.328	15.232	17.136	19 04	20 944	22 848	24 752	28 560
41	4.002	9 604	12.006	14 007	16 008	18.009	20 00	22 011	24.012	26 013	30 015
42	4 198	10 065	12.594	14 693	16 792	18.901	20 90	23.089	25 188	27.287	31 485
43	4 40	10 56	13 20	15 40	17.60	19 80	22 00	24 20	26 40	28 60	33 00
44	4 606	11 046	13 818	16 121	18 424	20 727	23 03	25.333	27 636	29 939	34 545
45	4 818	11 563	14.454	16 863	19 272	21 681	24 09	26 399	28 908	31 317	36 135
46	5.043	12.086	15.128	17.626	20 144	22.662	25.18	27 608	30 216	32 754	37.770
47	5 256	12.614	15.768	18.396	21.024	23.652	26 28	28 908	31.536	34 164	39 420
48	5 482	12 846	16 446	19.187	21 928	24 669	27.41	30 151	32 152	35.633	41.115
49	5 714	12 913	17.142	19.999	22.856	25 713	28 57	31.427	34 284	37 141	42 855
50	5.950	14 28	17 850	20 825	23.80	26.775	29 75	32 725	35.70	38 675	44.625
51	6.180	14 832	18 540	21.665	24 76	27 855	30 95	34 045	37.08	40 205	46 425
52	6.432	15.437	19.296	22 512	25.728	28 944	32 16	35 376	38 592	41.808	48 240
53	6.684	16.041	20.052	23.304	26 736	30 078	33.42	36 762	40 104	43 446	50 130
54	6.940	16 656	20 820	24 029	27 760	31.230	34.70	38 17	41 64	45.11	52.05
55	7 198	17 275	21 594	25.193	28.792	32.391	35 99	39.589	43.188	46 787	53 985
56	7 462	17 909	22.386	26.117	29.848	33.579	37 31	41.041	44.772	48.503	55.965
57	7 732	18.557	23.196	27 062	30 928	34.704	38.66	42.526	46 392	50 258	57.99
58	8 006	19.214	24 018	28.021	32.024	36.027	40 03	44.033	48 036	52 039	60.045
59	8.284	19.902	24.852	28 964	33.136	37 278	41.42	45.562	48.704	53 846	62.13
60	8 566	20 558	25 698	29 981	34.264	38.547	42.83	47 113	51.396	55.679	64.245

TABLE 5. — RELATIVE HUMIDITY, PER CENT — FAHRENHEIT TEMPERATURES

(Chap XXIII, p 954)

Pressure = 30 0 inches

Air Temp t	Depression of Wet bulb Thermometer (t − t′) 2	4	6	8	1 0	1 2	1 4	1 6	1 8	2 0	2 2	2 4	2 6	2 8	3 0	3 2	3 4	3 6	3 8	4 0
−40	46																			
−39	48																			
−38	50	2																		
−37	53	6																		
−36	56	10																		
−35	59	15																		
−34	61	20																		
−33	63	24																		
−32	64	28																		
−31	66	32	0																	
−30	68	36	4																	
−29	70	41	9																	
−28	72	45	15																	
−27	74	48	19																	
−26	75	51	24	0																
−25	76	53	28	5																
−24	77	55	32	10																
−23	78	57	36	15																
−22	80	59	39	20	0															
−21	81	61	43	24	5															
−20	82	63	45	28	10															
−19	83	65	48	32	15															
−18	84	67	51	35	19	2														
−17	85	69	53	39	23	7														
−16	86	70	56	42	27	12														
−15	86	72	58	45	31	17	4													
−14	87	74	61	48	34	21	8													
−13	88	75	63	50	38	25	13	0												
−12	88	76	64	52	41	29	17	6												
−11	89	77	66	55	44	32	21	10												
−10	90	78	68	57	46	36	25	14	4											
− 9	90	79	70	59	49	39	29	18	9											
− 8	90	81	71	61	51	42	32	22	13	3										
− 7	91	82	72	63	54	44	35	26	17	8										
− 6	91	82	73	64	56	47	38	29	20	12	3									
− 5	91	83	75	66	58	49	41	32	24	16	7									
− 4	92	84	76	68	60	52	44	36	28	20	12	4								
− 3	92	85	77	69	61	54	46	39	31	23	16	8	1							
− 2	92	85	78	71	63	56	49	42	34	27	19	12	5							
− 1	93	86	79	72	65	58	51	44	37	30	23	16	10	3						
0	93	87	80	73	67	60	53	47	40	33	27	20	14	7	1					
+ 1	93	87	81	75	68	62	56	49	43	36	30	24	18	11	5					
2	94	88	82	76	70	64	58	52	46	39	33	27	21	15	9	3				
3	94	88	82	77	71	65	59	54	48	42	36	30	25	19	13	7	2			
4	94	89	83	78	72	66	61	55	50	44	39	33	28	22	17	11	6	0		
5	95	89	84	78	73	68	63	57	52	46	41	36	31	25	20	15	10	4		
6	95	90	84	79	74	69	64	59	54	49	43	38	33	28	23	18	13	8	3	
7	95	90	85	80	75	70	65	60	55	51	46	41	36	31	26	21	17	12	7	2
8	95	90	86	81	76	71	67	62	57	53	48	43	38	34	29	24	20	15	11	6
9	95	91	86	82	77	72	68	63	59	55	50	46	41	36	32	27	23	18	14	10
10	96	91	87	82	78	73	69	65	60	56	52	47	43	39	34	30	26	22	17	13
11	96	91	87	83	79	74	70	66	62	58	53	49	45	41	37	33	28	25	20	16
12	96	92	88	84	80	75	71	67	63	59	55	51	47	43	39	35	31	27	23	19
13	96	92	88	84	80	76	73	69	65	61	57	53	49	45	41	38	34	30	26	23
14	96	92	89	85	81	77	74	70	66	62	59	55	51	48	44	40	37	33	29	26
15	96	93	89	86	82	78	75	71	67	64	60	57	53	50	46	42	39	35	32	29
16	96	93	90	86	82	79	76	72	69	65	62	58	55	51	48	45	41	38	34	31
17	97	93	90	86	83	80	77	73	70	66	63	60	57	53	50	47	43	40	37	34
18	97	93	90	87	84	80	77	74	71	68	65	61	58	55	52	49	45	42	39	36
19	97	94	90	87	84	81	78	75	72	69	66	63	60	56	53	50	47	44	41	38
20	97	94	91	88	85	82	79	76	73	70	67	64	61	58	55	52	49	46	43	40

t	(t − t′) 4 2	4 4	4 6	4 8	5 0	5 2	5 4	5 6	5 8	6 0
8	1									
9	5	1								
10	9	5	0							
11	1	8	4	0						
12	16	12	8	4	0					
13	19	15	11	7	4	0				
14	22	18	15	11	8	4	0			
15	25	21	18	14	11	7	4	0		
16	28	24	21	18	14	11	8	4	1	
17	30	27	24	21	17	14	11	8	5	1
18	33	30	27	24	20	17	14	11	8	5
19	35	32	29	26	23	20	17	14	11	8
20	37	34	32	29	26	23	20	17	14	1

t	(t − t′) 0 1	0 2	0 3	0 4	0 5
−50	50				
−49	54	5			
−48		12			
47	60	1			
−46	63	2			
−45	66	28			
−44	68	32			
−43	70	36	3		
−42	71	40	9		
−41	72	43	14		
−40	73	46	18		
−39	74	48	22		
−38	75	50	25	2	
−37	76	53	28	6	
−36	77	56	33	10	
−35	78	59	37	15	
−34	80	61	41	20	0
−33	81	63	44	24	5
−32	82	64	46	28	10
−31	83	66	49	32	15
−30	84	68	52	36	20

TABLE 5. — *Continued.*

Air Temp. *t*	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0	10.5
	Depression of Wet-bulb Thermometer (*t* − *t′*).																				
20	92	85	77	70	62	55	48	40	33	26	19	12	5								
21	92	85	78	71	63	56	49	42	35	28	21	15	8	1							
22	93	86	78	71	65	58	51	44	37	31	24	17	11	4							
23	93	86	79	72	66	59	52	46	39	33	26	20	14	7	1						
24	93	87	80	73	67	60	54	47	41	35	29	22	16	10	4						
25	94	87	81	74	68	62	55	49	43	37	31	25	19	13	7	1					
26	94	87	81	75	69	63	57	51	45	39	33	27	21	16	10	4					
27	94	88	82	76	70	64	58	52	47	41	35	29	24	18	13	7	2				
28	94	88	82	76	71	65	59	54	48	43	37	32	26	21	15	10	5				
29	94	88	83	77	72	66	60	55	50	44	39	34	28	23	18	13	8	3			
30	94	89	83	78	73	67	62	56	51	46	41	36	31	26	21	16	11	6	1		
31	94	89	84	78	73	68	63	58	52	47	42	37	33	28	23	18	13	8	4		
32	95	89	84	79	74	69	64	59	54	49	44	39	35	30	25	20	16	11	7	2	
33	95	90	85	80	75	70	65	60	56	51	46	41	37	32	27	23	18	14	9	5	0
34	95	90	86	81	76	71	66	62	57	52	48	43	38	34	29	25	21	16	12	8	3
35	95	91	86	81	77	72	67	63	58	54	49	45	40	36	32	27	23	19	14	10	6
36	95	91	86	82	77	73	68	64	60	55	51	46	42	38	34	29	25	21	17	13	9
37	95	91	87	83	78	74	69	65	61	57	53	48	44	40	36	31	27	23	19	15	11
38	96	91	87	83	79	75	70	66	62	58	54	50	46	42	37	33	29	25	21	17	14
39	96	92	87	83	79	75	71	67	63	59	55	51	47	43	39	35	31	27	24	20	16
40	96	92	87	83	79	75	71	68	64	60	56	52	48	45	41	37	33	29	26	22	18
41	96	92	88	84	80	76	72	69	65	61	57	54	50	46	42	39	35	31	28	24	20
42	96	92	88	85	81	77	73	69	65	62	58	55	51	47	44	40	36	33	30	26	23
43	96	92	88	85	81	77	73	70	66	63	59	55	52	48	45	42	38	35	31	28	25
44	96	93	89	85	81	78	74	71	67	63	60	56	53	49	46	43	39	36	33	30	26
45	96	93	89	86	82	78	74	71	67	64	61	57	54	51	47	44	41	38	34	31	28
46	96	93	89	86	82	79	75	72	68	65	61	58	55	52	48	45	42	39	35	32	29
47	96	93	89	86	82	79	75	72	69	66	62	59	56	53	49	46	43	40	37	34	31
48	96	93	90	86	83	79	76	73	69	66	63	60	57	54	50	47	44	41	38	35	32
49	96	93	90	86	83	80	76	73	70	67	64	61	57	54	51	48	45	42	39	36	34
50	96	93	90	87	83	80	77	74	71	67	64	61	58	55	52	49	46	43	41	38	35
51	97	94	90	87	84	81	78	75	71	68	65	62	59	56	53	50	47	45	42	39	36
52	97	94	90	87	84	81	78	75	72	69	66	63	60	57	54	51	49	46	43	40	37
53	97	94	90	87	84	81	78	75	72	69	66	63	61	58	55	52	50	47	44	41	39
54	97	94	91	88	85	82	79	76	73	70	67	64	61	59	56	53	50	48	45	42	40
55	97	94	91	88	85	82	79	76	73	70	68	65	62	59	57	54	51	49	46	43	41
56	97	94	91	88	85	82	79	76	73	71	68	65	63	60	57	55	52	50	47	44	42
57	97	94	91	88	85	82	80	77	74	71	69	66	63	61	58	55	53	50	48	45	43
58	97	94	91	88	85	83	80	77	74	72	69	66	64	61	59	56	54	51	49	46	44
59	97	94	91	89	86	83	80	78	75	72	70	67	65	62	59	57	55	52	49	47	45
60	97	94	91	89	86	83	81	78	75	73	70	68	65	63	60	58	55	53	50	48	46
61	97	94	92	89	86	84	81	78	76	73	71	68	65	63	61	58	56	54	51	49	47
62	97	94	92	89	86	84	81	79	76	74	71	69	66	64	61	59	57	54	52	50	47
63	97	95	92	89	87	84	82	79	77	74	71	69	67	64	62	60	57	55	53	50	48
64	97	95	92	90	87	84	82	79	77	74	72	70	67	65	63	60	58	56	53	51	49
65	97	95	92	90	87	85	82	80	77	75	72	70	68	66	63	61	59	56	54	52	50
66	97	95	92	90	87	85	82	80	78	75	73	71	68	66	64	61	59	57	55	53	51
67	97	95	92	90	87	85	83	80	78	75	73	71	69	66	64	62	60	58	56	53	51
68	97	95	92	90	88	85	83	80	78	76	74	71	69	67	65	62	60	58	56	54	52
69	97	95	93	90	88	85	83	81	79	76	74	72	70	67	65	63	61	59	57	55	53
70	98	95	93	90	88	86	83	81	79	77	74	72	70	68	66	64	61	59	57	55	53
71	98	95	93	90	88	86	84	81	79	77	75	72	70	68	66	64	62	60	58	56	54
72	98	95	93	91	88	86	84	82	79	77	75	73	71	69	67	65	63	61	59	57	55
73	98	95	93	91	88	86	84	82	80	78	75	73	71	69	67	65	63	61	59	57	55
74	98	95	93	91	89	86	84	82	80	78	76	74	71	69	67	65	63	61	60	58	56
75	98	96	93	91	89	86	84	82	80	78	76	74	72	70	68	66	64	62	60	58	56
76	98	96	93	91	89	87	84	82	80	78	76	74	72	70	68	66	64	62	61	59	57
77	98	96	93	91	89	87	85	83	81	79	77	74	72	71	69	67	65	63	61	59	57
78	98	96	93	91	89	87	85	83	81	79	77	75	73	71	69	67	65	63	62	60	58
79	98	96	93	91	89	87	85	83	81	79	77	75	73	71	69	68	66	64	62	60	58
80	98	96	94	91	89	87	85	83	81	79	77	75	74	72	70	68	66	64	62	61	59

TABLE 5. — *Continued.*

Air Temp t	Depression of Wet-bulb Thermometer $(t - t')$.																				
	11.0	11.5	12.0	12.5	13.0	13.5	14.0	14.5	15.0	15.5	16.0	16.5	17.0	17.5	18.0	18.5	19.0	19.5	20.0	20.5	21.0
35	2																				
36	5	1																			
37	7	3																			
38	10	6	2																		
39	12	8	5	1																	
40	15	11	7	4	0																
41	17	13	10	6	3																
42	19	16	12	9	5	2															
43	21	18	14	11	8	4	1														
44	23	20	16	13	10	7	4	0													
45	25	22	18	15	12	9	6	3													
46	26	23	20	17	14	11	8	5	2												
47	28	25	22	19	16	13	10	7	5	2											
48	29	26	23	21	18	15	12	9	7	4	1										
49	31	28	25	22	19	17	14	11	9	6	3	1									
50	32	29	27	24	21	18	16	13	10	8	5	3	0								
51	34	31	28	26	23	20	17	15	12	9	7	4	2								
52	35	32	29	27	24	22	19	17	14	11	9	6	4	1							
53	36	33	31	28	26	23	20	18	16	13	10	8	6	3	1						
54	37	35	32	29	27	24	22	20	17	15	12	10	8	5	3	1					
55	38	36	33	31	28	26	23	21	19	16	14	12	9	7	5	2	0				
56	39	37	34	32	30	27	25	22	20	18	16	13	11	9	7	4	2				
57	40	38	35	33	31	28	26	24	22	19	17	15	13	10	8	6	4	2			
58	41	39	37	34	32	30	27	25	23	21	18	16	14	12	10	8	6	3	1		
59	42	40	38	35	33	31	29	26	24	22	20	18	16	13	11	9	7	5	3	1	
60	43	41	39	37	34	32	30	28	26	23	21	19	17	15	13	11	9	7	5	3	1
61	44	42	40	38	35	33	31	29	27	25	22	20	18	16	14	12	10	8	7	5	3
62	45	43	41	39	36	34	32	30	28	26	24	22	20	18	16	14	12	10	8	6	4
63	46	44	42	40	37	35	33	31	29	27	25	23	21	19	17	15	13	11	10	8	6
64	47	45	43	41	38	36	34	32	30	28	26	24	22	20	18	17	15	13	11	9	7
65	48	46	44	41	39	37	35	33	31	29	27	25	24	22	20	18	16	14	12	11	9
66	48	46	44	42	40	38	36	34	32	30	29	27	25	23	21	19	17	16	14	12	10
67	49	47	45	43	41	39	37	35	33	31	30	28	26	24	22	20	19	17	15	13	12
68	50	48	46	44	42	40	38	36	34	32	31	29	27	25	23	21	20	18	16	15	13
69	51	49	47	45	43	41	39	37	35	33	32	30	28	26	24	23	21	19	18	16	14
70	51	49	48	46	44	42	40	38	36	34	33	31	29	27	25	24	22	20	19	17	15
71	52	50	48	46	45	43	41	39	37	35	33	32	30	28	27	25	23	22	20	18	17
72	53	51	49	47	45	43	42	40	38	36	34	33	31	29	28	26	24	23	21	19	18
73	53	51	50	48	46	44	42	40	39	37	35	34	32	30	29	27	25	24	22	20	19
74	54	52	50	48	47	45	43	41	39	38	36	34	33	31	29	28	26	25	23	21	20
75	54	53	51	49	47	45	44	42	40	39	37	35	34	32	30	29	27	26	24	23	21
76	55	53	51	50	48	46	44	43	41	39	38	36	34	33	31	30	28	27	25	24	22
77	56	54	52	50	48	47	45	43	42	40	39	37	35	34	32	31	29	28	26	25	23
78	56	54	53	51	49	47	46	44	43	41	39	38	36	34	33	31	30	28	27	25	24
79	57	55	53	51	50	48	46	45	43	42	40	38	37	35	34	32	31	29	28	26	25
80	57	55	54	52	50	49	47	45	44	42	41	39	38	36	35	33	32	30	29	27	26

t	$(t - t')$														
	21.5	22.0	22.5	23.0	23.5	24.0	24.5	25.0	25.5	26.0	26.5	27.0	27.5	28.0	28.5
61	1														
62	2	1													
63	4	2	0												
64	6	4	2	0											
65	7	5	4	2	0										
66	9	7	5	3	2	0									
67	10	8	7	5	3	2									
68	11	10	8	6	5	3	1								
69	13	11	9	8	6	5	3	1							
70	14	12	11	9	8	6	4	3	1						
71	15	13	12	10	9	7	6	4	3	1					
72	16	15	13	12	10	9	7	6	4	3	1				
73	17	16	14	13	11	10	8	7	5	4	3	1			
74	18	17	15	14	13	11	10	8	7	5	4	3	1		
75	20	18	17	15	14	12	11	9	8	7	5	4	3	1	
76	21	19	18	16	15	13	12	11	9	8	6	5	4	3	1
77	22	20	19	17	16	14	13	12	10	9	8	6	5	4	3
78	23	21	20	18	17	16	14	13	11	10	9	8	6	5	4
79	23	22	21	19	18	17	15	14	13	11	10	9	7	6	5
80	24	23	22	20	19	18	16	15	14	12	11	10	9	7	6

TABLE 5. — *Continued.*

Air Temp. t	Depression of Wet-bulb Thermometer ($t - t'$).														
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44
82	96	92	88	84	80	76	72	69	65	61	58	55	51	48	45
84	96	92	88	84	80	76	73	69	66	62	59	56	52	49	46
86	96	92	88	84	81	77	73	70	66	63	60	57	53	50	47
88	96	92	88	85	81	77	74	70	67	64	61	57	54	51	48
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49
92	96	92	89	85	82	78	75	72	68	65	62	59	56	53	50
94	96	93	89	85	82	79	75	72	69	66	63	60	57	54	51
96	96	93	89	86	82	79	76	73	69	66	63	61	58	55	52
98	96	93	89	86	83	79	76	73	70	67	64	61	58	56	53
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54
102	96	93	90	86	83	80	77	74	71	68	65	62	60	57	55
104	97	93	90	87	83	80	77	74	71	69	66	63	60	58	55
106	97	93	90	87	84	81	78	75	72	70	66	64	61	58	56
108	97	93	90	87	84	81	78	75	72	70	67	64	62	59	57
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60	57
112	97	94	90	87	84	81	79	76	73	70	68	65	63	60	58
114	97	94	91	88	85	82	79	76	74	71	68	66	63	61	58
116	97	94	91	88	85	82	79	76	74	71	69	66	64	61	59
118	97	94	91	88	85	82	79	77	74	72	69	67	64	62	59
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62	60
122	97	94	91	88	85	83	80	77	75	72	70	67	65	63	60
124	97	94	91	88	85	83	80	78	75	73	70	68	65	63	61
126	97	94	91	88	86	83	80	78	75	73	70	68	66	64	61
128	97	94	91	89	86	83	81	78	76	73	71	68	66	64	62
130	97	94	91	89	86	83	81	78	76	73	71	69	67	64	62
132	97	94	92	89	86	84	81	79	76	74	71	69	67	65	63
134	97	94	92	89	86	84	81	79	76	74	72	69	67	65	63
136	97	94	92	89	86	84	81	79	77	74	72	70	68	65	63
138	97	94	92	89	87	84	82	79	77	75	72	70	68	66	64
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66	64

t	Depression of Wet-bulb Thermometer ($t - t'$)														
	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
80	41	38	35	32	29	26	23	20	18	15	12	10	7	5	3
82	42	39	36	33	30	28	25	22	20	17	14	12	10	7	5
84	43	40	37	35	32	29	26	24	21	19	16	14	12	9	7
86	44	42	39	36	33	31	28	26	23	21	18	16	14	11	9
88	46	43	40	37	35	32	30	27	25	22	20	18	15	13	11
90	47	44	41	39	36	34	31	29	26	24	22	19	17	15	13
92	48	45	42	40	37	35	32	30	28	25	23	21	19	17	15
94	49	46	43	41	38	36	33	31	29	27	24	22	20	18	16
96	50	47	44	42	39	37	35	32	30	28	26	24	22	20	18
98	50	48	45	43	40	38	36	34	32	29	27	25	23	21	19
100	51	49	46	44	41	39	37	35	33	30	28	26	24	22	21
102	52	49	47	45	42	40	38	36	34	32	30	28	26	24	22
104	53	50	48	46	43	41	39	37	35	33	31	29	27	25	23
106	53	51	49	46	44	42	40	38	36	34	32	30	28	26	24
108	54	52	49	47	45	43	41	39	37	35	33	31	29	27	25
110	55	52	50	48	46	44	42	40	38	36	34	32	30	28	26
112	55	53	51	49	47	44	42	40	38	36	35	33	31	29	27
114	56	54	52	49	47	45	43	41	39	37	35	34	32	30	28
116	57	54	52	50	48	46	44	42	40	38	36	34	33	31	29
118	57	55	53	51	49	47	45	43	41	39	37	35	34	32	30
120	58	55	53	51	49	47	45	43	41	40	38	36	34	33	31
122	58	56	54	52	50	48	46	44	42	40	39	37	35	34	32
124	59	57	54	52	50	48	47	45	43	41	39	38	36	34	33
126	59	57	55	53	51	49	47	45	44	42	40	38	37	35	33
128	60	58	56	54	52	50	48	46	44	42	41	39	37	36	34
130	60	58	56	54	52	50	48	47	45	43	41	40	38	37	35
132	61	58	56	55	53	51	49	47	45	44	42	40	39	37	36
134	61	59	57	55	53	51	49	48	46	44	43	41	39	38	36
136	61	59	57	55	54	52	50	48	46	45	43	41	40	38	37
138	62	60	58	56	54	52	50	49	47	45	44	42	40	39	37
140	62	60	58	56	54	53	51	49	47	46	44	43	41	40	38

TABLE 5 — *Concluded*

Air Temp *t*	Depression of Wet bulb Thermometer (*t* − *t′*)														
	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45
80	0														
82	2	0													
84	5	3	0												
86	7	5	3	1											
88	9	7	5	3	1										
90	11	9	7	5	3	1									
92	13	11	9	7	5	3	1								
94	14	12	10	9	7	5	3	1							
96	16	14	12	10	8	7	5	3	2	0					
98	17	15	14	12	10	8	7	5	3	[illegible]	0				
100	19	17	15	13	12	10	8	7	5	4	2	1			
102	20	18	16	15	13	11	10	8	7	5	4		1		
104	21	20	18	16	14	13	11	10	8	7	5	4	2	1	
106	23	21	19	17	16	14	13	11	10	8	7	5	4	3	1
108	24	22	20	19	17	16	14	1	11	10	8	7	5	4	3
110	25	23	21	20	18	17	15	14	12	11	10	8	7	6	4
112	26	24	23	21	19	18	16	15	14	12	11	9	8	7	6
114	27	25	24	22	20	19	18	16	15	13	12	11	9	8	7
116	28	26	25	23	22	20	19	17	16	14	13	12	11	9	8
118	29	27	25	24	23	21	20	18	17	16	14	13	12	11	9
120	29	28	26	25	23	22	21	19	18	17	15	14	13	12	10
122	30	29	27	26	24	23	22	20	19	18	16	15	14	13	11
124	31	30	28	27	25	24	22	21	20	18	17	16	15	14	12
126	32	30	29	27	26	25	23	2	1	19	18	17	16	15	13
128	33	31	30	28	27	25	24	23	22	0	19	18	17	16	14
130	33	32	30	29	28	26	25	24	22	21	20	19	18	16	15
132	34	33	31	30	28	27	26	24	23	22	21	20	18	17	16
134	35	33	32	30	29	28	26	25	24	23	21	20	19	18	17
136	35	34	33	31	30	28	27	26	25	23	22	21	20	19	18
138	36	35	33	32	30	29	28	27	25	24	23	22	21	20	19
140	37	35	34	32	31	30	29	27	26	25	24	3	1	20	19

t	Depression of Wet bulb Thermometer (*t* − *t′*)														
	46	47	48	49	50	51	5	53	54	55	56	57	58	59	60
106	0														
108	2	0													
110	3	2	1												
112	4	3	2	1											
114	6	5	3	2	1										
116	7	6	5	4	2	1	0								
118	8	7	6	5	4	3	2	1							
120	9	8	7	6	5	4	3	2	1						
122	10	9	8	7	6	5	4	3	2	1	0				
124	11	10	9	8	7	6	5	4	3	2	1	0			
126	12	11	10	9	8	7	6	5	4	3	2	2	1		
128	13	12	11	10	9	8	7	6	5	4	4	3	2	1	0
130	14	13	12	11	10	9	8	7	6	5	5	4	3	2	1
132	15	14	13	12	11	10	9	8	7	6	6	5	4	3	2
134	16	15	14	13	12	11	10	9	8	7	6	6	5	4	3
136	17	16	15	14	13	12	11	10	9	8	7	6	6	5	4
138	17	16	15	14	14	13	12	11	10	9	8	7	7	6	5
140	18	17	16	15	14	13	12	12	11	10	9	8	7	7	6

TABLE 6. — THEORETICAL WATER-RATE COMPUTATION TABLE FOR STEAM ENGINES.

(Chap. XVI, p. 656) (Thompson's Table)

T. P.	0	1	2	3	4	5	6	7	8	9
3	117 300	121.015	124 717	128 406	132.083	135 748	139 399	143.075	146 665	150.279
4	153.880	157 514	161 137	164 750	168 353	171 945	175 527	179.098	182 659	186.210
5	189 750	193 336	196 914	200 483	204 044	207 598	211 142	214 679	218.208	221.728
6	225.240	228.799	232 351	235 897	239 437	242 970	246 497	250 017	253 531	257 039
7	260.540	264.056	267 506	271 071	274 570	278 063	281 550	285 031	288 506	291.976
8	295 440	298 922	302 400	305 872	309 338	312 800	316 256	319 708	323.154	326 594
9	330 030	333 488	336 941	340 389	343 833	247 273	350 707	354 137	357 563	360 984
10	364 400	367 842	371 280	374 714	378 144	381 570	384 992	388 410	391 824	395.234
11	398 640	402.064	405 485	408 902	412 315	415 725	419 131	422 534	425.933	429 328
12	432.720	436.120	439 517	442 911	446 301	449 688	453 071	456 451	459.828	463 200
13	466 570	469 950	473 326	476 699	480 068	483 435	486 798	490.159	493 516	496.869
14	500 220	503 596	506 968	510 338	513 706	517 070	520 432	523.790	527.146	530.500
15	533 850	537 213	540 573	543 930	547 285	550 638	553 987	557 334	560 679	564.011
16	567.360	570 713	574 063	577 411	580 757	584.100	587 441	590.780	594 115	597 449
17	600 780	604 109	607 435	610 759	614 081	617 400	620 717	624 031	627 343	630.653
18	633 960	637.265	640 567	643.867	647.165	650 460	653 753	657.043	660.331	663 617
19	666 900	670 200	673 498	676 793	680 086	683 378	686 666	689.953	693 238	696.520
20	699 800	703 098	706 394	709 688	712.980	716 270	719 558	722.844	726 128	729 410
21	732 690	735 968	739 244	742 518	745 790	749 060	752 328	755 504	758 858	762 120
22	765 380	768 660	771 938	775 215	778 490	781 763	785 034	788 303	791.570	794 836
23	798 100	801 362	804 622	807 881	811.138	814 393	817.646	820.897	824 146	827 334
24	830.640	833 908	837 175	840 440	843 703	846 965	850 225	853 484	856 741	859 996
25	863 250	866.502	869 753	873.002	876 249	879.495	882 739	885 983	889 223	892 462
26	895 700	898 936	902 171	905 404	908.635	911.865	915 093	918.320	921.545	924.768
27	927 990	931.210	934 429	937.646	940 831	944 075	947 287	950 498	953.707	956.914
28	960 120	963.352	966.583	969 813	973 011	976 268	979.493	982 717	985 939	989.160
29	992 380	995.598	998 815	1002 031	1005.245	1008 458	1011.669	1014.879	1018 087	1021 294
30	1024.500	1027 704	1030 907	1034 109	1037.309	1040 508	1043 705	1046.901	1050 095	1053.288
31	1056.480	1059 670	1062 859	1066 047	1069 233	1072 418	1075 601	1078 783	1081 963	1085.142

TABLE 6 — *Concluded.*

T. P.	0	1	2	3	4	5	6	7	8	9
32	1088.320	1091.528	1094.736	1097 942	1101 146	1104.350	1107 552	1110.754	1113.954	1117.152
33	1020.350	1123 546	1126 742	1129 936	1133.128	1136 420	1139 510	1142.700	1145.888	1149.074
34	1152.260	1155.444	1158.628	1161.810	1164 990	1168.170	1171 348	1174.526	1177.702	1180.876
35	1184 050	1187.222	1190 394	1193.564	1196.732	1199.900	1203 066	1206.232	1209 396	1212.558
36	1215 720	1218 917	1222.112	1225.307	1228 500	1231.693	1234.884	1238.075	1241 264	1244.453
37	1247 640	1250 827	1254 012	1257 107	1260.380	1263.563	1266 744	1269 925	1273.104	1276.283
38	1279 460	1282.637	1285 812	1288.987	1292 160	1295.333	1298 504	1301.675	1304.844	1308.013
39	1311 180	1314.347	1317 512	1320.677	1323 840	1327.003	1330 164	1333.325	1336.484	1339.643
40	1342 800	1345.957	1349.112	1352.267	1355 420	1358 573	1371.724	1364.875	1368.024	1371.173
41	1374.320	1377.467	1380 612	1383 757	1386 900	1390 043	1393.184	1396.325	1399.464	1402.603
42	1405 740	1408 877	1412.012	1415 147	1418 280	1421.413	1424.544	1427.675	1430.804	1433.933
43	1437.060	1440 230	1443 398	1446 506	1449 734	1452 900	1456.066	1459.230	1462.394	1465.558
44	1468 720	1471.882	1475.042	1478.202	1481 362	1484 520	1487.678	1490.834	1493.990	1497.146
45	1500.300	1503.454	1506.606	1509.758	1512 910	1516 060	1519.210	1522.359	1525.506	1528.654
46	1531.800	1534 946	1538 090	1541 234	1544 378	1547 520	1550 662	1553 802	1556.942	1560.082
47	1563 220	1566 358	1569.494	1572.630	1575 766	1578.900	1582.034	1585.166	1588.298	1591.430
48	1594.560	1597 690	1600 818	1603 946	1607.074	1610.200	1613 326	1616 450	1619.574	1622.698
49	1625 820	1628.942	1632 062	1635 182	1638 302	1641.420	1644.538	1647.654	1650.770	1653.886
50	1657.000	1660.114	1663.226	1666.338	1669.450	1672.560	1675.670	1678 778	1681.886	1684.994
51	1688.100	1691.206	1694.310	1607 414	1700.518	1703 620	1706.722	1709.822	1712.922	1716.022
52	1719.120	1722.218	1725 314	1728 410	1731.506	1734 600	1737.604	1740 786	1743.878	1746.970
53	1750.060	1753.150	1756.238	1759 327	1762.414	1765 500	1768 586	1771.670	1774.754	1777.838
54	1780.920	1784.002	1787.082	1790 162	1793 242	1796 320	1799.398	1802 474	1805.550	1808 626
55	1811.700	1814.829	1817 957	1821.084	1824 211	1827 338	1830 463	1833 588	1836.713	1839.837
56	1842.960	1846.083	1849.205	1852 326	1855 447	1858 568	1861.687	1864 806	1867.925	1871.043
57	1874 160	1877.277	1880 393	1883.508	1886 623	1889 738	1892.851	1895.964	1899.077	1902 189
58	1905.300	1908.411	1911.521	1914.630	1917 739	1920 848	1923.955	1927 062	1930.169	1933.275
59	1936.380	1939.485	1942 589	1945 692	1948.795	1951.898	1954.999	1958 100	1961.201	1964 301
60	1967.400	1970.499	1973.597	1976.694	1979.791	1982 888	1985.983	1989 078	1992.173	1995 267

TABLE 7. — PROPERTIES OF AMMONIA (NH_3).

(Chap. XXIV, p. 997.)

Abs. Press. Lbs. per sq. in.	Fahr. Temp.	q	r	λ or H	ϕ_L	ϕ_V	ϕ_{SAT}	Sp. Vol. Sat. NH_3
10	−47.5	−83 5	584.30	500.80	−.1816	1.4145	1 2329	21 50
15	−27.5	−62.95	572.25	509.50	−.1355	1.3240	1.1885	17.00
20	−17 0	−51 50	565 75	514 25	−.1105	1.2772	1 1667	13.50
25	− 8 5	−42 50	560 50	518.00	−.0800	1 2412	1.1512	10.35
30	0 0	−33 75	556 00	522.25	− 0706	1.2070	1 1364	9.00
35	+6.25	−27.10	551.63	524 53	−.0562	1.1830	1.1268	7.87
40	12.0	−21.00	548 00	527 00	−.0433	1.1610	1 1177	6.92
45	17.2	−15 65	544 90	529.25	−.0322	1.1420	1 1098	6.20
50	21.85	−10 62	542 00	531.38	− 0220	1 1245	1 1025	5.60
55	26.35	− 5 80	539.35	533.55	−.0117	1 1075	1.0952	5.15
60	30 30	− 1 73	536.78	535 05	− 0028	1 0940	1.0912	4 75
65	34 20	+ 2 40	534 30	536.70	+.0060	1.0807	1.0867	4.40
70	38.00	6.40	531 88	538 28	.0147	1 0680	1.0827	4.10
75	41 3	9 80	529 81	539 61	.0215	1 0565	1 0780	3 85
80	44 4	13 00	527 85	540.85	.0278	1 0460	1 0738	3.6
85	47.6	16 18	525 83	542 01	0342	1 0350	1.0692	3 4
90	50 6	19 40	523 95	543 35	0403	1.0250	1 0653	3 22
95	53 4	22 45	522 15	544 60	0459	1 0160	1 0619	3.04
100	56 1	25 30	520 40	545 70	0514	1 0080	1 0594	2.90
105	58 8	28 10	518 70	546 80	.0570	9997	1.0567	2 75
110	61 3	30 80	517 10	547 90	0619	.9920	1 0539	2.63
115	63.6	33 20	515 60	548 80	0663	9842	1 0505	2 50
120	65.9	35 55	514 12	549 67	.0708	9766	1 0474	2 41
125	68 15	37 95	512 67	550 62	.0752	9690	1 0442	2.32
130	70.45	40 40	511 25	551 65	.0798	.9630	1 0428	2 25
135	72 5	42 55	509.90	552 45	0837	9571	1 0408	2 15
140	74 5	44 65	508 60	553 25	.0875	.9515	1.0390	2 07
145	76.52	46 77	507 32	554 09	0915	.9457	1.0372	2.00
150	78.52	48.88	506 05	554.93	.0952	9402	1.0354	1 95
155	80 42	50.88	504.85	555.73	.0988	9348	1 0336	1 90
160	82 15	52 68	503 70	556 38	1025	9292	1 0317	1 85
165	83 93	54 55	502 50	557 05	1061	9234	1 0295	1.80
170	85.70	56.45	501 35	557 80	.1096	9183	1 0279	1 75
175	87 05	58 22	500 28	558 50	1130	9146	1 0273	1 70
180	89 12	60 05	499 15	559 20	1163	9103	1 0266	1 65
185	90 85	61 90	498 05	559.95	1195	9043	1 0238	1 61
190	92.35	63 40	497 05	560 45	1222	9009	1 0231	1 58
195	93 90	65 05	496 00	561 05	1250	8962	1.0212	1.54
200	95 45	66 75	494 97	561 72	.1279	8915	1 0194	1 50
205	96.18	68 30	493 95	562 25	.1306	.8870	1.0176	1 47
210	98.5	69 90	492 95	562.85	1333	.8825	1.0158	1 43
215	100 05	71 50	492.00	563.50	.1360	8780	1.0140	1.39

TABLE 8. — PROPERTIES OF SULPHUR DIOXIDE (SO_2).

(Chap. XXIV, p. 997.)

t	p	q	r	ϕ_L	ϕ_V	Sp. Vol.
−40	3.16	−23.3	174.1	−.047	.365	20.9
−30	4.5	−20.0	173.5	−.041	.362	16.0
−20	6.1	−16.8	172.5	−.034	.358	12.0
−10	8.0	−13.5	171.7	−.028	.354	9.2
0	10.3	−10.3	170.6	−.021	.350	7.2
10	13.3	−7.0	169.2	−.015	.346	5.0
20	17.0	−3.8	167.3	−.008	.341	4.6
30	21.5	−0.5	165.0	−.001	.335	3.7
40	27.1	2.6	162.0	.005	.330	3.0
50	33.5	5.9	158.8	.012	.324	2.44
60	41.0	9.2	155.0	.018	.318	2.0
70	50.0	12.5	151.3	.025	.312	1.7
80	60.0	15.7	147.0	.032	.305	1.4
90	72.0	19.0	142.3	.038	.298	1.1
100	84.5	22.2	137.5	.045	.290	0.9

TABLE 9. — PROPERTIES OF CARBON DIOXIDE (CO_2).

(Chap. XXIV, p. 997.)

t	p	q	r	ϕ_L	ϕ_V	Sp. Vol.
−30	185	−27.8	130.0	−.060	.240	.490
−20	225	−23.9	126.5	−.051	.235	.417
−10	265	−19.7	122.0	−.042	.230	.350
0	310	−15.2	117.8	−.032	.224	.294
10	360	−10.8	112.0	−.022	.218	.246
20	420	−5.9	106.5	−.012	.211	.206
30	490	−0.8	100.0	−.001	.204	.172
40	570	+4.6	93.0	.010	.197	.145
50	655	10.3	85.9	.021	.189	.120
60	750	16.8	76.0	.034	.180	.100
70	850	24.5	64.0	.048	.169	.081
80	965	35.0	46.0	.067	.153	.062
88.5	1070	59.3	0.0	.112	.112	.035

88.5 is critical temperature for CO_2.

TABLE 10. — AMMONIA ABSORPTION IN POUNDS OF AMMONIA ABSORBED PER POUND OF WATER.

Temperature ° F.	Pressure Pounds per Square Inch Absolute.					
	15	20	25	30	35	40
10	1.38	1 76	2 15	2.54	2.92	3 34
20	1 09	1.40	1.70	2 01	2 31	2.64
30	92	1 18	1 43	1.69	1 94	2.21
40	.80	1 01	1.23	1.46	1.67	1 91
50	.70	.89	1 08	1.38	1 48	1 68
60	.63	.80	96	1.14	1.31	1.49
70	56	.71	.86	1.02	1 17	1 34
80	.50	.64	78	92	1 06	1 20
90	.45	.57	.69	82	95	1 08
100	40	51	.62	73	84	.96
110	36	46	56	67	76	87
120	.33	41	50	60	69	.78
130	29	39	.45	54	.62	71
140	.26	33	.40	.48	55	.62
150	23	29	36	.42	.48	.55
160	20	26	31	.38	43	48
170	18	23	28	33	38	43
180	16	19	23	.28	33	37
190	13	17	20	.24	28	32
200	12	14	17	.20	23	.26

TABLE 11. — PROPERTIES OF NaCl BRINE.

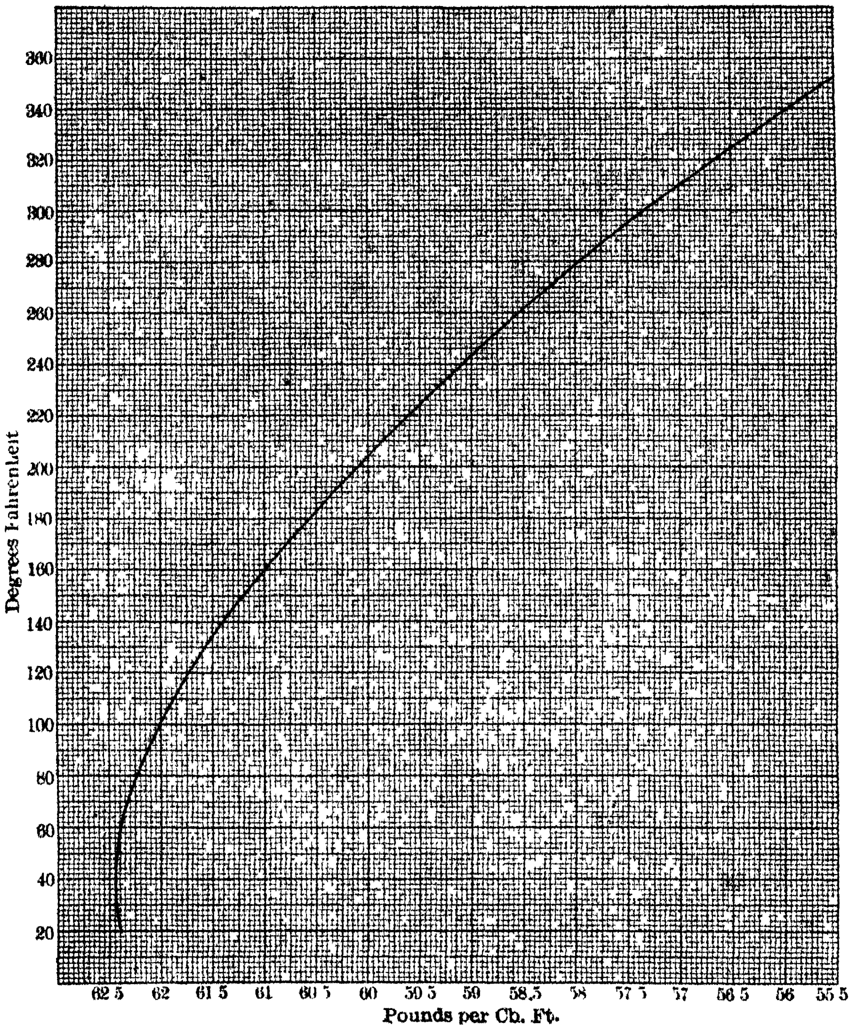
(Chap. XXIV, p. 1009.)

Per cent of the Salt by Weight	Specific Gravity at 39° F.	Degrees on Salometer at 60° F.	Freezing Point. ° F.	Specific Heat.
1	1 007	4	+30 5	992
2	1 015	8	+29.3	984
2 5	1 019	10	+28 6	980
3	1 023	12	+27.8	.976
3 5	1 026	14	+27 1	972
4	1 030	16	+26 6	968
5	1 037	20	+25 2	960
6	1 045	24	+23 9	946
7	1 053	28	+22 5	.932
8	1 061	32	+21 2	919
9	1 068	36	+19.9	905
10	1 076	40	+18 7	892
12	1 091	48	+16.0	.874
15	1 115	60	+12 2	855
20	1 155	80	+ 6 1	.829
24	1.187	96	+ 1.2	.795
25	1 196	100	+ 0 5	783
26	1 204	104	− 1.1	771

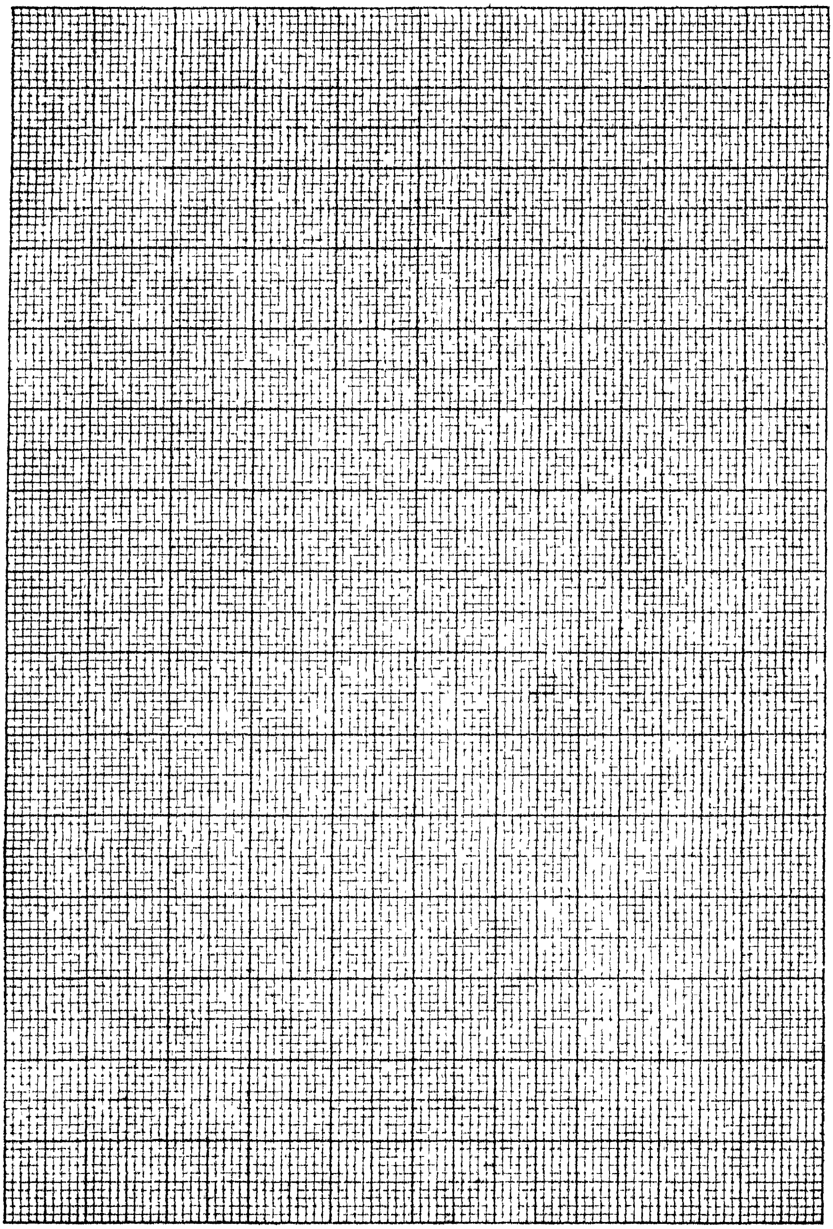
TABLE 12. — PROPERTIES OF $CaCl_2$ BRINE.

(Chap. XXIV, p. 1009.)

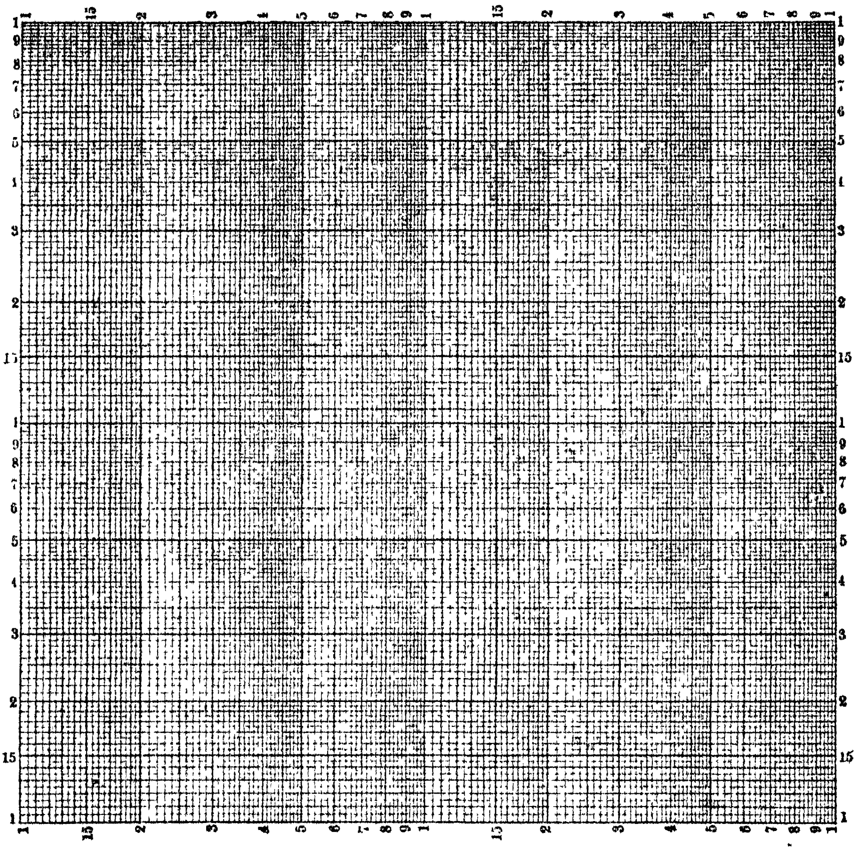
Per cent of the Salt by Weight	Specific Gravity at 60° F.	Degrees on Salometer at 60° F.	Freezing Point. ° F.	Specific Heat.
1	1.007	4	+31.10	.996
2	1.015	8	+30.38	.988
3	1.024	12	+29.48	.980
4	1.032	16	+28.58	.972
5	1.041	22	+27.68	.964
6	1.049	26	+26.60	.960
7	1.058	32	+25.52	.936
8	1.067	36	+24.26	.925
9	1.076	40	+22.8	.911
10	1.085	44	+21.3	.896
11	1.094	48	+19.7	.890
12	1.103	52	+18.1	.884
13	1.112	58	+16.3	.876
14	1.121	62	+14.3	.868
15	1.131	68	+12.2	.860
16	1.140	72	+10.0	.854
17	1.150	76	+ 7.5	.849
18	1.159	80	+ 4.6	.844
19	1.169	84	+ 1.7	.839
20	1.179	88	− 1.4	.834
21	1.189	92	− 4.9	.825
22	1.199	96	− 8.6	.817
23	1.209	100	−11.6	.808
24	1.219	104	−17.1	.799
25	1.229	108	−21.8	.790
26	1.240	112	−27.0	.778
27	1.250	116	−32.6	.760
28	1.261	120	−39.2	.757
29	1.272		−46.3	
30	1.283		−54.4	
31	1.294		−52.5	



Curve Showing Weight of Water in Pounds per Cubic Foot for Temperatures from 20–350° F



Cross-section Paper.



LOGARITHMIC CROSS-SECTION PAPER.

INDEX

ABRASION test, 135.
Absorbing power of reagents in flue gas analysis, 520.
Absorption dynamometers, 276.
  Alden brake, 284.
  Brakes employing friction of liquids, 286.
  Prony brakes of different types, 276–280.
Absorption system of refrigeration, 999.
Accelerated test for cement, 163.
Acid tests for oils, 245.
Acoustic current meter, 403.
Adiabatic compression of dry air, 956.
Adulteration of oils, 243.
Air compressing machinery, types of, 938.
Air compression, adiabatic, 956.
Air compression, combined diagram for multistage, 969.
Air compression, efficiency of, 965.
Air compression, efficiency in multistage, 972.
Air compression, intercooler, 967.
Air compression, isothermal, 955.
Air compression, slip, 963.
Air compression, temperature rise, 957.
Air compression, work of, 954.
Air compressor, cooling of, 939.
Air compressor, effect of clearance in, 963.
Air compressor horse power, 961.
Air compressor, hydraulic, 947.
Air compressor, mechanical efficiency of, 966.
Air compressor, multistage, 967.
Air compressors, piston, 938.
  Dry, 939.
  Water injection, 939.
  Wet, 939.
Air compressor, Rateau turbine, 946.
Air compressors, rotary or positive, 942.
Air compressor tests (see testing of air compressors).
Air compressor, turbine, 945.
Air, dry, characteristics of, 949.
Air, dry, work of compression, 955.
Air horse power, 966.
Air horse power, centrifugal fan, 983.
Air, humidity of, 950.
Air lift, 1084.
Air machines, for refrigeration, 998.
Air, moist, characteristics of, 950.
Air, moist, work of compression of, 958.
Air, physical characteristics of, 949.
Air required for combustion of fuel gases 858–860.
Air thermometer, 199.
Air thermometer, corrections for, 199.
Air thermometer, directions for use of, 200.
Alcohol fuel, 861.
Alden brake, 284.
American Boiler Manufacturers' Association specifications, 143.
American Foundrymen's Association, standard specifications for testing gray cast iron, 149.
American Society for Testing Materials, specifications for cement, 163.
American Society of Civil Engineers and American Society for Testing Materials, standard method of testing cement, 158.
American Waterworks Association, standard specifications for cast-iron water pipe, 151.
Ammonia, absorption in water, table, 1109.
Ammonia compression, wet and dry, 1021.
Ammonia compressor, 1004.
Ammonia compressor indicator diagram, 1021.
Ammonia condensers, 1007.
Ammonia, properties of, table, 1107.
Ammonia, weight of in circulation, 1024.
Amsler polar planimeter, 21.
Analysis of coal,
  Ash, 505.
  Fixed carbon, 504.
  Moisture, 502.
  Volatile matter, 502.
Analysis of exhaust gases, 877.
Analysis of fuel gases, 868.
Analysis of fuels, 500.
Anemometer, 439.
Anemometer calibration, 440.
Anemometers, use of, 441.
Angular deformation, definition of, 51.
Approximate formula for flow of steam through orifices, 442.
Approximate method of determining chemical analysis of coal on basis of proximate analysis, 507.

Apron method of determining velocity in channel, 399.
Arndt econometer, 528.
Ashcroft oil testing machine, 267.
Aspirator for flue gas sampling, 514–516.
Atomic weight of gases, 855.
Atwater bomb calorimeter, 474.
Autographic apparatus, combined with extensometer, 107.
Autographic diagrams, 14.
Autographic tachometers, 23.

BARREL calorimeter, 550.
Barrel calorimeter, directions for use, 552.
Barrus continuous calorimeter, 556.
Barrus superheating calorimeter, 558.
Base metal thermocouples, 213.
Bauschinger extensometer, 100.
Beam formulas, 60.
Beaumé hydrometer, 246.
Beaumé hydrometer scale with corresponding specific gravities, table of, 1091.
Belt dynamometer, 306.
Belt testing machine, 318.
Belt tests (see testing of belts).
Bending test, 135.
Blower, centrifugal or volume, 943.
Blower, rotary or positive, 942.
Blower tests (see testing of centrifugal fans).
Bomb calorimeter, 471.
Bomb calorimeter, calculations for, 480–483.
Bomb calorimeter, directions for use, 478.
Boston micrometer extensometer, 106.
Boult's lubricant testing machine, 268.
Bourdon gauge, 181.
Boyle's law, 327.
Brakes,
  Alden, 284.
  Electric generator absorption dynamometer, 291.
  Electro-magnetic, 290.
  Employing friction of liquids, 286.
  Fan, 289.
  Pump, 288.
  Prony, 276–280.
  Self-regulating, 283
  Westinghouse, 287.
Brauer's method for measuring continuous flow by means of orifices, 367.
Brayton cycle, 349.
Breaking load, 53.
Breast wheel, 1047.
Brine used for cooling, properties of, 1009.
Brine used for refrigerating purposes, table of properties of, 1109, 1110.
Brinell ball test for hardness, 136.
Bristol pyrometer, 214.
Bristol recording gauge, 187.
Burnham steam meters, 454.
Burning point in oils, definition of, 256.
Burning point in oils, method of determination of, 256.
Busby extensometer, 102.

CALCULATING machines,
  Brunsviga, 49.
  Fuller rule, 19.
  Grant, 49.
  Slide rule, 16.
  Thatcher rule, 18
Calculation, approximate, rules and formulæ for, 8.
Calculations, numerical, accuracy of, 11.
Calibration of anemometers, 440.
Calibration of current meters, 404.
Calibration of gas meters, 437.
Calibration of indicator springs, 597–605
Calibration of mercury thermometers, 220
  Film or stem correction, 221.
  Test for boiling point, 220.
  Test for freezing point, 221.
  Test for other temperatures, 221.
Calibration of Morin dynamometer, 295.
Calibration of nozzles, 380.
Calibration of orifices, 365.
Calibration of Pitot tubes, 409.
Calibration of planimeters, 37.
Calibration of pressure gauges, 188–191.
Calibration of pyrometers, 222.
  Optical and radiation pyrometers, 224.
  Resistance thermometers, 223.
  Thermocouples, 222.
Calibration of thermometers and pyrometers, 219
Calibration of Venturi meter, 380.
Calibration of water meters, 396.
Calibration of Webber dynamometer, 301.
Calibration of weirs, 380.
Calorific power, 466.
Calorimetric methods of engine testing, 799.
Calorimetric pyrometers, 206.
Calorimetry, fuel, 470.
Capacity test of steam engine, 726.
Capacity water meters, 389.
Carbon dioxide, properties of, table, 1108.
Carburetors, 846.
Carnot cycle, 347, 351, 352.
Carnot efficiency of steam engines, 729.
Carpenter coal calorimeter, calculations for, 489.
Carpenter fuel calorimeter, 485.
Cathetometer, 47.
Cement test, accelerated or hot, 163.
Cement test, normal, 163.

Cement testing (see testing of cement).
Cement testing machines, 96.
Center vent turbine, 1059.
Centrifugal blowers, 943.
Centrifugal fan, action of, 981.
Centrifugal pump, 1068.
Centrifugal pump, testing of (see testing of centrifugal pumps).
Centrifugal pump, theoretical lift or head, 1074, 1075.
Centrifugal pump, theoretical power, 1074.
Centrifugal pump, theory of action, 1068-1071.
Centrifugal pumps, classification of, 1071-1074.
Change of entropy at constant pressure in gases, 333.
Change of entropy at constant volume in gases, 332.
Change of entropy in gases, general case, 333.
Charles's Law, 327.
Chemical or ultimate analysis of coal, 501.
Chemical steam calorimeter, 575.
Chill point, determination of, 252.
Chronograph, 232.
Cistern manometer, 171.
Clearance determined from indicator diagram, 652.
Clearance, effect of in air compressors, 963.
Clausius cycle, 352, 353.
Coefficient of contraction, 361.
Coal analysis, 501.
Coefficient of discharge for orifices used for measuring gas, 424-429.
Coefficient of discharge for steam in orifice measurements, 449-452.
Coefficient of discharge for Venturi meter, Herschel's experiments, 371.
Coefficients of discharge for water through mouthpieces, 364.
Coefficient of efflux or discharge, 361.
Coefficient of friction, belt transmission, 323.
Coefficient of friction, flow of water in pipes, 414.
Coefficient of friction for oils, measurement of, 256.
Coefficient of friction in oils, 236.
Coefficient of friction in belt drives, 323.
Coefficients of friction, table of, 1090.
Coefficients of heat transmission, table of, 1011.
Coefficient of performance, ideal, 1014.
Coefficient of velocity, 361.
Coffin planimeter, 31.
Coffin planimeter, theory of, 31.
Cold gas efficiency of gas producer, 903.
Combination gas producer, 855.
Combined diagram for multicylinder steam engine, construction of, 661-670.
Combined diagram for two-stage air compressor, 969-972.
Combustion, 465.
Combustion, air required for, 858.
Combustion calculations, 512.
Combustion, heat of, 466.
Combustion, losses in, 531.
Combustion of elementary fuels, 466.
Combustion of fuels containing hydrogen, 467.
Combustion of hydrogen, 467.
Combustion, temperatures attained in, 468.
Combustion, weight and volume relations, 465.
Composition of fuels, 498.
Compressed air, pumping by, 1084.
Compression loading, formulas for, 56.
Compression micrometers,
  Olsen, 112.
  Riehlé, 111.
Compression of ammonia, wet and dry, 1021.
Compression of dry air, 955.
Compression of moist air, 958.
Compression refrigeration plant, 1004.
Compression test machines (see testing machines).
Compression tests, directions for, 125.
Compression tests, form of test piece, 125.
Compressor horse power, 966.
Computations, aids to, 48.
Condensing calorimeter, forms of, 550.
Constancy of volume in cements, 163.
Constant-pressure engines, 841.
Constant-volume engines, 841.
Continuous jet calorimeter, 552.
Continuous speed counter, 228.
Contraction in weirs, 373.
Cooling effect of refrigeration, 1028.
Cooling of air compressors, 939.
Cooling systems in refrigeration, 1007.
Counting devices, 227.
Cradle dynamometers, 314.
Critical pressure for steam in orifice measurements, 448.
Critical pressure in orifice and nozzle measurements with gases, 421.
Crosby gauge testing apparatus, 191.
Crosby indicator, 586.
Cross section paper, logarithmic, 14.
Crude oil, 860.
Current meters, 402.
Current water meters, 387.
Curtis turbine, 812.
Cycles and their efficiencies, 346.
Cycles for steam, 351.
  Carnot, for saturated steam, 351.

Cycles, Carnot, for wet, dry, or superheated vapor, 352.
  Clausius, for saturated steam, 352.
  Clausius, for superheated steam, 353.
  Rankine, for saturated steam, 354.
  Rankine, for superheated steam, 355.
Cylinder condensation, 657.
Cylinder efficiency of steam engines, 729.

Dead weight calibration of indicator springs, 597.
Deflection in transverse loading, 58.
Deflectometer, 111.
Deformation, definition of, 51.
DeLaval steam turbine, 810.
Denny & Johnson torsion meter, 313.
Density of gases, 855.
Density of oils, method of determination, 247.
  By weighing, 247.
  With hydrometer, 247.
Depth of water over weirs, 372.
Determination of average velocity in streams, 404.
Determination of radiation rate for bomb calorimeter, 476, 482.
Dew point apparatus, 952.
Diagram, stress-deformation, 53.
Diagram water rate, 653.
Diagram water rate, approximate formulas for, 656.
Diagram water rate for multicylinder engine, 655.
Diagrams, autographic, 14.
Diaphragm gauge, 182.
Diesel cycle, 350.
Differential dynamometer, 300.
Diffuser, in fan blowers, 943.
Dilution coefficient flue gas analysis, 538.
Directions for compression testing, 125.
Directions for taking indicator diagrams, 636
Directions for tension tests, 118.
Directions for torsion testing, 129–131.
Directions for transverse testing, 127.
Direct shear, 61.
Direct shear tests, 132.
Disc fan, 943.
Displaced diagrams and time diagrams, 676.
Doble impulse wheel, 1049.
Dosch smoke observer, 542.
Double zone producer, 854.
Down draft producer, 854.
Draft gauges 175.
Drifting test, 135.
Drop tests, 94, 133.
Dry air, characteristics of, 949.
Dry gas meters, 434.
Ductility, definition of, 52.
DuLong's formula, 511.
Durability of lubricants, 266.
Durability testing machines, for oil, 267.
Durability tests of oils, directions for, 270.
Durand dynamometer, 306.
Duty of pumping engine, 775.
Dynamometer car, locomotive tests, 795
Dynamometer, traction, 292.
Dynamometer, transmission, 293.
  Belt, 306.
  Denny & Johnson, 313.
  Differential or Webber, 300.
  Durand, 306.
  Emerson power scale, 304.
  Föttinger, 314.
  General types, 293.
  Kenerson, 310.
  Lewis, 298.
  Morin, 293.
  Pillow block, 297.
  Steel yard, 296.
  Torsion, 309, 312.
  Van Winkle, 305.

Eckhardt steam meter, 457.
Economy test of steam engine, 726.
Edson recording gauge, 185.
Efficiency in multistage air compression, 972.
Efficiency standard for locomotives, 787.
Efficiency standards for steam engine, 728,
Efficiency of actual refrigeration process. 1019.
Efficiency of air compression,
  Mechanical, 966.
  Volumetric, 963, 964.
Efficiency of belt transmission, 323.
Efficiency of centrifugal fans,
  Manometric, 985.
  Mechanical, 986.
  Volumetric, 986.
Efficiency of centrifugal pumps, 1074.
  Actual, 1077.
  Hydraulic, 1076.
  Manometric, 1076.
Efficiency of cooler in refrigeration. 1020.
Efficiency of gas engine,
  Cylinder, 921.
  Mechanical, 923.
  Thermal, 922.
  Volumetric, 924.
Efficiency of gas producers, 903.
Efficiency of hydraulic ram, 1039.
Efficiency of impulse wheel, 1052.
  Actual or gross, 1052.
  Hydraulic, 1052.
Efficiency of injector, 833.
Efficiency of injector, as a pump, 834.
  Heat, 835.

Efficiency of insulation, 1012.
Efficiency of jet pump, 1043.
Efficiency of pulsometer, 1082.
Efficiency of refrigeration process, 1033.
Efficiency of steam engine,
  Carnot, 729.
  Cylinder, 729.
  Mechanical, 730.
  Thermal, 729.
  Thermodynamic, 729.
Efficiency of steam turbines,
  Cylinder, 821.
  Potential or Clausius, 821.
  Thermal, 820.
Efficiency of water-pressure engines, 1044.
Efficiency of water turbines, 1057, 1062.
Efficiency of water wheels, 1046–1048.
Efficiency, mechanical, of ammonia plant, 1033.
Efficiency, plant, 730.
Efficiency tests, 3.
Ejector, 832.
Elastic curve in transverse loading, 58.
Elastic limit, definition of, 52.
Elastic limit strength, definition of, 52.
Elasticity, modulus of, 52.
Electric generator absorption dynamometer, 291
Electric meters for measuring gas, 444.
Electric pyrometer, compensating device for low resistance type, 215.
Electric pyrometer, high resistance type, 212.
Electric pyrometer, low resistance type, 213.
Electrical measurement of power input and output, 315.
Electrical methods of measuring temperature, 209.
Electromagnetic brake, 290.
Elementary fuels, combustion of, 466.
Ellison universal steam calorimeter, 578.
Elongation, equivalent in tension tests, 123.
Emerson bomb calorimeter, 475.
Emerson power scale, 304.
Emery material testing machine, 72, 78, 82.
Empirical formulas, deduction of, 7.
Engineering or proximate analysis of coal 50.
Engines, water pressure, 1043.
Entropy diagram for ammonia, 996.
Entropy diagram for gas engine, 685.
Entropy diagram for steam engine, 670–675.
Entropy of gases, 332.
Entropy of vapors, 340.
Equation of state or condition for ideal gas, 327.
Equivalent elongation in tension tests, 123.
Ericsson cycle, 348.
Ericsson engine, 926.
Errors, accidental, 4.
Errors, combination of, 7.
Errors, experimental, classification of, 4.
Errors in different types of planimeters, 40.
Errors in engine indicators, 595.
Errors in engine indicators, discussion of effects, 612–620.
Errors in flue gas analysis, 530, 531, 538.
Errors in weighing, 360.
Errors in weir measurement, effects of, 380.
Errors of single observations, 5.
Errors, mean, 5.
Errors, neglect of, 10.
Errors, probability of, 5.
Errors, systematic, 4.
Ertel current meter, 402.
Evaluation of indicator diagrams taken with faulty springs, 608–612.
Evaporation, factor of, 722–724.
Excess air, computation for, in gas engine tests, 880.
Excess coefficient, flue gas analysis, 538.
Exhaust gas analysis, 877.
Exhaust, heat loss in, 877–883.
Exhaust loss, experimental determination of, 883.
Expansion and fusion pyrometers, 205.
Expansion, ratio of, 645, 748.
Experiments, classification of, 3.
Explosion recorder, 909.
Extensometer,
  Bauschinger, 100.
  Boston, 106.
  Busby, 102.
  Combined with autographic apparatus, 107.
  Henning, 104.
  Johnson, 103
  Kenerson autographic, 110.
  Marshall, 105.
  Olsen, 106.
  Olsen wire, 107.
  Payne, 102.
  Strohmeyer, 101.
  Thurston, 103.
Extensometer, forms of, 100.
Extensometer and autographic apparatus combined, 107.
External latent heat, definition of, 334.

FACTOR of evaporation, 722–724.
Factor of safety, definition of, 53.
Fairbanks automatic cement tester, 96.
Fan, air horse power, 983.
Fan blowers, 943–945.

Fan blower, diffuser, 943.
Fan brakes, 289.
Fan, centrifugal, action of, 981.
Fan, disc, 943.
Fan, head produced by, actual, 982.
Fan, head produced by, theoretical, 981.
Fan tests (see testing of centrifugal fans).
Fan, work done by, 983.
Fatigue test, 137.
Favre & Silbermann fuel calorimeter, 483.
Fery absorption pyrometer, 216.
Fery radiation pyrometer, 217.
Figure of merit, gas producer tests, 904.
Fineness of cement, 159.
Flash point of oil, method of determination, 254, 255.
  With open cup, 254.
  With closed cup, 255.
Float measurement of velocity, 397.
Flow of air, formulas for, 432.
Flow of steam, Grashof's formula, 452.
Flow of steam in pipes, 463.
Flow of steam, Napier's formula, 453.
Flow of steam through orifices, approximate formulas for, 442.
Flow of water in pipes, 412.
Flow of water over weirs, 372.
Flue gas analysis, 512, 517.
Flue gas analysis, accuracy of, 530.
Flue gas analysis apparatus, automatic types, 527.
Flue gas analysis apparatus, general forms, 520.
Flue gas analysis apparatus, method of operation, 520.
Flue gas analysis,
  Determination of CO_2, 517.
  Determination of carbon monoxide, 518.
  Determination of hydrogen, 519.
  Determination of moisture, 519.
  Determination of oxygen, 518.
Flue gas analysis, errors in, 530, 531, 538.
Flue gas analysis, reagents used, 517, 520.
Flue gas analysis, sampling, 513.
Flue gas apparatus,
  Elliott, 522.
  Hempel, 525.
  Orsat, 524
Flue loss in boilers, computation of, 531-538.
Flue losses, classification of, 531.
Fluid friction in gas engines, definition of, 678.
Fluid pressure methods of calibrating indicator springs, 602.
Forging test, 135.
Formulas for beams, 60.
Formulas for compression loading for long columns, 56.
Formulas of compression loading for short columns, 56.
Formulas, empirical, deduction of, 7.
Formulas for tension loading, 55.
Formulas for torsion loading, 61.
Formulas for transverse loading, 57.
Fourneyron turbine, 1056.
Four-cycle gas engines, 843.
Föttinger torsion meter, 314.
Frahm's resonance tachometer, 231.
Francis turbine, 1059.
Free evaporation, 333.
Friction brakes, 276-288.
Friction, classification of, 237.
Friction of motion, 237.
Friction of rest, 237.
Friction,
  Angle of repose, 236.
  Coefficient of friction, 236.
  Definitions, 236.
  Normal force, 236.
  Total pressure, 236.
Friction horse power in gas engines, 908
Friction of ball and roller bearings, 271.
Friction of cords and belts, 238.
Friction of fluids, 241.
Friction of pivots and collars, 238.
Fuel analysis, 500.
Fuel calorimeter, continuous type, 471.
Fuel calorimeter, discontinuous type, 471.
Fuel calorimetry, 470.
Fuel gas, analysis of, 868.
Fuel gas analysis, apparatus for, 871-875.
Fuel gases, constants for, 858.
  Air required for combustion, 858.
  Heating value, 858.
Fuels, classification of, 499.
Fuels, commercial, 498
Fuels, gas engine, 846
Fuller's slide rule, 19.

Gas analysis,
  Heinz apparatus, 871.
  Siebert-Kühn apparatus, 875.
Gas cycles, 347.
  Brayton, 349.
  Carnot, 347.
  Diesel, 350
  Ericsson, 348.
  Otto, 349.
  Stirling, 348.
Gas engine,
  Constant pressure, 841.
  Constant volume, 841.
  Cylinder arrangement, 846.
  Four-cycle, 843.
  Two-cycle, 844.
Gas engine diagrams, discussion of, 678-685.

Gas engine diagrams, inertia effects in, 684.
Gas engine diagram, lower loop, 678, 680.
Gas engine diagram, pump, 678.
Gas engine diagrams, abnormal features, 680–685.
Gas engine, fluid friction loss in, 678.
Gas engine, friction horse power, 908.
Gas engine fuels, 846.
Gas engine, heat balance, 920.
Gas engine, indicated horse power, 901.
Gas engine testing (see testing of gas engines).
Gas meters,
  Dry, 434.
  Turbine, 436.
  Wet, 432.
Gas meter calibration, 437.
Gas meter capacities, 437.
Gas producer,
  Combination plant, 853.
  Double zone, 854.
  Down draft, 854.
  Pressure, 851.
  Series, 854.
  Suction, 851.
Gas producer efficiency, 903
Gas producer, figure of merit, 904
Gas producer, heat balance, 899–903.
Gas producer testing (see testing of gas producers).
Gas producer, types of, 850.
Gas thermometer, theory of, 196
Gases,
  Atomic weight, 855
  Density, 855.
  Specific weight, 855
Gases, hydrocarbon, table of contents for, 858.
Gases, pressure and volume changes,
  Adiabatic, 330.
  Constant pressure, 329.
  Constant volume, 330.
  Isentropic, 330.
  Isothermal, 330.
  Pv^n = constant, 331.
Gases, table of constants for, 857.
Gases, table of mean specific heats, 469.
Gasometer, 430.
Gauge calibration, 188.
  By comparison with mercury column, 189.
  By comparison with other gauges, 188.
  By comparison with standard weights, 190.
Gauges, corrections for, 192.
Gauge syphons, 185.
Gauges, pressure,
  Bourdon, 181.
  Diaphragm, 182.
Gauges (*continued*),
  Draft, 175, 181.
  Recording, 185, 186, 187.
  Vacuum, 185.
Generators used as brakes, 317.
Graphical representation of experimental results, 12.
Grashof's formula, flow of steam, 452.
Gumming or drying tests for oils, 245.
  Bailey's apparatus, 245.
  Nasmyth's apparatus, 245.
HAMMER test, 135.
Hammond water weigher, 384.
Hand speed counter, 227.
Hardening test, 135.
Hardness test, 136.
Head of water over weirs, 372.
Head on weirs, measurement of, 378.
Head pumped against, centrifugal pump, 1074, 1075.
Head pumped against, steam pump, 774.
Heat analysis, 799.
Heat analysis applied to multicylinder engines, 806.
Heat balance for boiler, 721.
Heat balance for gas engine, 920.
Heat balance for gas producer, 899–903.
Heat balance for hot-air engine, 935.
Heat balance for steam engine, 731.
Heat chart, 337.
Heat diagram for steam, use of, 337.
Heat efficiency of injector, 835.
Heat lost in exhaust of gas engine, computation of, 877–883.
Heat of combustion, 466.
Heat of the liquid, definition of, 334.
Heat transmission, coefficients of, table, 1011
Heat unit, 325.
Heating value, fuel gases, 858.
Heating value of fuels, by computation, 510.
Heating value of fuels, determination of, 505.
Heinz apparatus for gas analysis, 871.
Heisler impact testing machine, 94.
Hempel fuel calorimeter, 473.
Henning extensometer, 104.
Hersey rotary meter, 390.
Hersey torrent meter, 387.
Hess testing machine for friction of ball and roller bearings, 272.
Higher and lower heating value of fuels, 467.
Hirn's analysis, 799–806.
Hoadley calorimetric pyrometer, 208.
Hoadley draft gauge, 181.
Hoadley steam calorimeter, 554.
Holborn pyrometer, 216.
Hook gauge, 378.
Hopkinson optical indicator, 640.

Horizontal and vertical shear in transverse loading, 59.
Horse power, definition of, 276.
Hot-air engines, methods of operation, 925.
Hot-air engines tests (see testing of hot-air engines).
Hot-air engines, types of, 925.
Hot gas efficiency of gas producer, 904.
Hot-water meters, 394.
Humidity, absolute, 951.
Humidity chart, 954
Humidity, determination of, 951.
Humidity of air, definition of, 950.
Humidity, relative, 951.
Humidity tables, 1100–1104.
Humphrey internal combustion pump, 1082.
Hydraulic compressor, 947.
Hydraulic machinery, classification of, 1037.
Hydraulic ram, 1038.
Hydraulic ram, efficiency, 1039.
Hydraulic ram tests (see testing of hydraulic rams).
Hydrodynamometer, 400.
Hydrometric pendulum, 400.
Hyperbola, method of drawing, 648.

ICE machine (see refrigeration).
Ice machines, testing of (see testing of vapor compression refrigerating machines).
Ice making, 1008
Ice making capacity, 1028.
Ice melting capacity, 1028.
Impact tests, 132.
Impact testing machine, 94.
Impulse type of steam turbine, 808.
Impulse water turbine, 1043.
Impulse water wheel, 1049.
Impulse wheel, Doble, 1046.
Impulse wheel, theory of action of water on buckets, 1050–1052.
Impurities in oils, 243.
Indicated horse power, 647.
Indicated horse power, formula for, 582.
Indicated horse power, gas engines, 907.
Indicated horse power, hot-air engines, 933.
Indicator diagram (see also steam engine indicator diagram).
Indicator diagram, ammonia compressor, 1021.
Indicator diagram, directions for taking, 636.
Indicator diagram, displaced and time, 676.
Indicator diagrams, effects of bad steam connections, leaky valves and pistons, 660.
Indicator diagram, errors due to indicators and indicator connections, 658.
Indicator diagram, errors due to valve setting, 658.
Indicator diagram, evaluation of, 608–612.
Indicator diagram, steam chest and steam pipe, 675.
Indicator diagram for four-cycle engines, 678-680.
Indicator diagram for two-cycle engines, 685.
Indicator diagrams for gas engines, discussion of, 678–685.
Indicator diagrams, various forms, 657.
Indicators for ammonia machines, 592.
Indicator, optical, 637, 638, 640.
Indicator springs, calibration of, 605.
Indicator springs, forms of, 597.
Indicator springs, scale of, 608.
Indicator, steam engine (see steam engine indicator).
Inferential water meters, 387.
Injectors,
  Monitor, 832.
  Penberthy, 830.
  Schütte-Koerting, 831.
Injector blower, 946
Injector, general theory of, 827.
Injector, heat efficiency of, 855.
Injector, pump efficiency of, 834.
Injector tests, 835.
Injector tests, report, 837–840.
Injectors, construction and operation, 829.
Insulation, 1010.
Insulation, efficiency of, 1011.
Insulation, testing of, 1011.
Intercooler, in air compression, 967.
Internal horse power, steam turbine, 819.
Internal latent heat, definition of, 334.
Isentropic or adiabatic curve for steam, 650.
Isothermal compression of dry air, 955.

JET pump, 1041.
Jet pump efficiency, 1043.
Johnson extensometer, 103.
Jonval turbine, 1050.
Junker gas calorimeter, 493.

KENERSON, autographic extensometer, 110.
Kenerson dynamometer, 310.
Kerr turbine, 816.

LATENT heat of vaporization, definition of, 334.
Laws of friction, 242.
Laws of vapors, 333.

Leakage test for boilers, 712.
Leakage test for pumping engines, 783.
LeChatelier electric pyrometer, 212.
Lewis dynamometer, 298.
Liquid fuels, characteristics of, 860.
Limits of separating calorimeter, 573.
Limits of throttling calorimeter, 564.
Liquid pressure pyrometer, 202.
Lloyd's specifications for quality and testing of ship steel, 145.
Locomotives, testing of (see standard method of conducting efficiency tests of locomotives).
Logarithmic cross section paper, 14.
Lord's apparatus for tar and soot determination, 888.
Losses in pipe fittings and valves, flow of water, 413-416.
Losses in steam turbines, 822.
Lower loop diagram for gas engines, 678, 680.
Low pressure water meters, 386.

MAHLER bomb calorimeter, 472.
Manograph, 638.
Manometer, 168.
Manometers,
  Cistern, 171.
  Differential, 174.
  Multiplying, 174.
  U-shaped, 169.
Mathot explosion recorder, 909.
Manufacturers' standard specifications for structural cast iron, 141.
Manufacturers' standard specifications for special open hearth plate and rivet steel, 141.
Manufacturers' standard specifications for structural steel, 139.
Marking gauges, tension tests, 119.
Marshall extensometer, 105.
Maximum load, 53.
Mean ordinate by polar planimeter, 29.
Mean specific heat for gases, 469.
Measurement of average velocity in channels and streams, 397.
Measurement of compression, in materials, 111.
Measurement of elongation, in materials, 99.
Measurement of gas by calorimetry, 443.
Measurement of gas by determining average velocity, 439.
Measurement of gas by Pitot tube, 441.
Measurement of gas by Venturi meter, 442.
Measurement of gas by volume displacement, 430.
Measurement of gases, classification of methods, 418.
Measurement of gases by means of orifices, theoretical velocity attained, and weight discharged, 419.
Measurement of liquids by means of nozzles, 368.
Measurement of liquids, gases, and vapors, 356.
Measurement of pressure, 168.
Measurement of speed, 227.
Measurement of strain, 99.
Measurement of torsion and deflection, 111.
Measurement of vapors, classification of methods, 445.
Measurement of vapors by determining average velocity, 453.
Measurement of vapors by means of orifices, 446.
Measurement of vapors by weighing, 445.
Measurement of water by means of orifices, 361.
Measurement of water by weighing, 357.
Mechanical equivalent of heat, 325.
Mechanical refrigeration, theory of, 992.
Mercury and alcohol thermometers, 202.
Mercury columns, 172.
Mercury column, corrections for, 173.
Mercury thermometers, calibration of, 220.
Mercury thermometers, rules for care of, 204.
Mesuré & Nouel pyrometric telescope, 217.
Metallic pyrometer, 205.
Meter prover, 432.
Methods of testing materials, 115.
Methods of testing cement, 158.
Micrometer, 43.
Micrometer caliper, 44.
Micrometer screws, accuracy of, 43.
Minor tests for materials, 134.
Mixed flow turbine, 1059.
Modulus of elasticity, definition of, 52.
Modulus of rigidity, 61.
Modulus of resilience, definition of, 52, 61.
Modulus of rupture in torsion, 62.
Modulus of rupture in transverse loading, 59.
Modulus of shear elasticity, 61.
Moist air, characteristics of, 950.
Moisture in steam, direct determination of, 567.
Mollier diagram, 337.
Morin's rotation dynamometer, 293.
Morin dynamometer, calibration of, 295.
Morse thermo gauge, 216.
Mosscrop speed recorder, 230.
Motors used as transmission dynamometers, 316.

Multicylinder steam engines, heat analysis, 806.
Multiplying and differential manometers, 174.
Multistage compression of air, 967.
Multistage impulse turbine, 812.
Multistage reaction turbine, 815.

NAPIER's formula, flow of steam, 453.
Nash disc meter, 391.
Natural cement, 164.
Neutral loading in transverse loading, 58.
Normal consistency in cement testing, 159.
Normal stress, definition of, 51.
Normal tests for cement, 163.
Nozzle measurements for water, 368.

OBSERVATIONS, doubtful, rejection of, 10.
Oil testing machines,
  Ashcroft, 267.
  Boult, 268.
  Hess, 272.
  Olsen, 262.
  Rankine, 256, 257.
  Riehlé, 262.
  Thurston, 258–261.
Oil tests (see testing of oils).
Oils,
  Adulteration of, 243.
  Impurities in, 243.
Olsen cement testing machine, 99.
Olsen compression micrometer, 112.
Olsen-Cornell oil testing machine, 262.
Olsen material testing machines, 70, 86, 91, 99.
Olsen extensometer, 106.
Olsen new wire extensometer, 107.
Open type water meters, 386.
Optical and radiation pyrometer, 216.
Optical indicator, 637.
Orifice measurement for vapors, 446.
Orifice measurements for water, 361.
Orifice methods for measuring gas, precautions necessary in, 429.
Orifices, calibration of, 365.
Orifices in thin plates, coefficients of discharge, for water, 363.
Orifices for measuring continuous flow, 366.
Orifices, submerged, 361.
Otto cycle, 349.
Overshot water wheel, 1045.
Oxygen, preparation of, for fuel calorimetry, 473.

PANTOGRAPH reducing motion, 627–630.
Parallel flow turbine, 1059.
Parr fuel calorimeter, 490.
Payne extensometer, 102.
Pendulum reducing motion, 621–627.
Pennsylvania Railroad Company's specifications for cast-iron car wheels, 151.
Perfect engine, performance of, 730.
Perfect engine, steam consumption for, 731.
Performance, coefficients of, in refrigeration, 1014, 1019.
Pillow block dynamometer, 297.
Piston air compressors, 938.
Piston water meters, 393.
Pitot tube for closed conduits, 406.
Pitot tube for measuring gas, 441.
Pitot tube for open channels, 405.
Pitot tube, Gregory, 408.
Pitot tube, modified form of, 409.
Pitot tube, Taylor, 408.
Planimeter,
  Amsler, 21.
  Coffin, 31.
  Polar, 21.
  Roller, 35.
  Suspended, 31.
  Willis, 34.
Planimeter calibration, 37.
Planimeter, theory of instrument, 23.
Planimeter, the zero circle, 24.
Planimeters, directions for care and use of, 36.
Planimeters, errors in different types, 40.
Planimeters, special, 42.
Plant efficiency, 730.
Pneumatic displacement pump, 1080.
Polar planimeter, 21.
Polar planimeter, forms of, 28.
Portland cement, 165.
Positive water meters, 393.
Power and efficiency of overshot water wheel, 1046.
Power and efficiency of radial inward flow, parallel flow, and mixed flow turbines, 1062.
Power, definitions of, 275.
Power, indicated and dynamometric, 581.
Power, measurement of, 276.
Power of water turbines, 1057, 1062.
Power of water wheels, 1046.
Power output of turbines, 819.
Power required by centrifugal pumps 1074.
Pressure, absolute, 167.
Pressure and volume changes in gases, 329.
Pressure and volume changes in vapors, 343.
Pressure, barometer, 167.
Pressure below atmosphere, 168.
Pressure gas producer, 851.
Pressure gauge, 181.

Pressure, gauge or manometer, 167.
Pressure, measurement of, 168.
Pressure stages in turbines, 812.
Pressure units, 167.
Primary stresses, definition of, 63.
Probability of errors, 5.
Producer (see gas producer).
Producer gas analysis (see gas analysis).
Producer gas, quantity made per pound of fuel, 884.
Producer gas, theory of manufacture, 847.
Prony brake, cooling and lubrication, 283.
Prony brake, design of, 280.
Prony brakes, 276–280.
Propeller fan, 943.
Properties of steam, tables of, 1092–1098.
Psychrometer, 952.
Pulsometer, 1080.
Pulsometer, efficiency, 1082.
Pump brake, 288.
Pumps, centrifugal, 1068.
Pump diagram for gas engines, 678.
Pump efficiency of injector, 834.
Pump horse power, hot-air engines, 933.
Pump, Humphrey internal combustion, 1082.
Pump, jet, 1041.
Pump, pneumatic displacement, 1086
Pump, reciprocating, 1067.
Pump, rotary, 1067.
Pump, turbine, 1068.
Pumping by compressed air, 1084.
Pumping engine, testing of (see standard method of conducting duty trials of pumping engines).
Punching test, 135.
Pyrometers,
  Bristol, 214.
  Calorimetric, 206.
  Electric, 212.
  Expansion, 205.
  Fery absorption, 216.
  Fery radiation, 217.
  Fusion, 205.
  Hoadley, 208.
  Holborn, 216.
  LeChatelier, 212.
  Liquid pressure, 202.
  Mesuré & Nouel, 217.
  Metallic, 205.
  Morse, 216.
  Optical, 216.
  Radiation, 216.
  Seger cones, 206.
  Vapor pressure, 201.
  Wanner, 216.
  Wedgewood, 205.

QUALITY curves on combined steam engine diagram, 666.
Quality determinations for steam, importance of, 544.
Quality of steam, definition of, 543.
Quality of steam, methods of determining, 544.

RADIAL inward flow turbine, 1059.
Radial outward flow reaction turbine, 1056.
Radial outward flow reaction turbine, power and efficiency of, 1057.
Radiation loss from steam engines, 805.
Radiation rate, determination of, for fuel calorimeter, 476, 482.
Rankine cycle, 354, 355.
Rankine's oil testing machine, dynamic apparatus, 257.
Rankine's oil testing machine, static apparatus, 256.
Ranson chronograph, 234.
Rateau turbine, 812.
Rateau turbine compressor, 946.
Ratio of expansion, 645, 748.
Reaction type of steam turbine, 808.
Reaction water turbine, 1053.
Reagents for flue gas analysis, 517–520.
Reciprocating pumps, 1067.
Recording gauges, 185.
Rectangular weir, theory of, 372.
Reducing motion for indicators, 621.
Reducing wheels, 630–632.
Reduction of area, definition of, 52.
Reëvaporation, 657.
Refrigerating agents, 994.
  Ammonia, 994.
  Carbonic acid, 994.
  Ethyl and methyl chloride, 994.
  Pictet fluid, 994.
  Sulphuric ether, 994.
  Sulphurous acid, 994.
Refrigerating effect, 1028.
Refrigerating machine, compressor horse power, 1033.
Refrigerating machine, simple type, 993.
Refrigerating machine,
  Air, 998.
  Ammonia absorption, 999.
  Vacuum process, 1001.
  Vapor compression, 1003.
Refrigerating machines, classification of, 997.
Refrigerating machines, capacity of, 1027.
Refrigerating machines, rating of, 1027.
Refrigeration cycle, actual for vapor compression machine, 1016.
Refrigeration, cooling effect, 1028.
Refrigeration, cooling systems, 1007.
Refrigeration cycle, ideal, for vapor compression machine, 1012.

Refrigeration, mechanical, theory of, 992.
Refrigeration process, actual efficiency of, 1019.
Refrigeration, tonnage, 1028.
Regulation tests for steam engines, 727.
Reservoir method of measuring gases, 437.
Resilience, modulus of, 52, 61.
Resistance thermometer, 210.
Riehlé cement testing machine, 99.
Riehlé compression micrometer, 111.
Riehlé material testing machines, 70, 84, 90, 90.
Riehlé oil testing machine, 262.
Rigidity, modulus of, 61.
Ringelmann's smoke chart, 541.
Risdon-Alcott turbine, 1060.
Road tests, locomotive, 786.
Rod floats, 398.
Roller planimeter, 35.
Rotary pumps, 1067.
Rotary pumps, testing of, 1077.
Rules for conducting boiler trials, Am. Soc. Mech. Eng'rs. Code of 1899, 692–707.
  Alternate method of starting and stopping, 695.
  Boiler efficiencies, 699.
  Calibration of apparatus, 693.
  Character of coal, 692.
  Duration of test, 694.
  Examination of boiler, 692.
  Flue gas analysis, 698.
  Fuel analysis, 698.
  Heat balance, 700.
  Keeping the records, 696.
  Preparation of boiler and connection, 693.
  Report, 701.
  Sampling coal, 697.
  Smoke observations, 699.
  Standard method of starting and stopping, 695.
  Starting and stopping test, 694.
  Treatment of ashes and refuse, 697.
  Uniformity of conditions, 695.
Rules for conducting steam engine tests, A.S.M.E. Code of 1902, 732.
  Calibration of instruments, 734.
  Coal used, 734.
  Complete engine and boiler test, 738.
  Condition of plant, 732.
  Duration of test, 737.
  Indicated horse power, 742.
  Indicated diagrams, 747–753.
  Measurement of coal, 741.
  Measurement of feed water, 740.
  Measurement of heat units used, 739.
  Object of test, 732.
Rules for conducting steam engine tests, A.S.M.E. Code of 1902 (*continued*),
  Quality of steam, 745.
  Report of tests, 755–773.
  Standards of economy and efficiency, 753.
  Standard heat test, 737.
  Starting and stopping, 737.
  Steam used by auxiliaries, 741.
  Uniformity of conditions, 747.

SALINOMETER, 1010.
Sampling of coal for analysis, 501.
Sarco carbon dioxide recorder, 529.
Saturation curve, ammonia compressor diagram, 1023.
Saturation curve for steam, construction of, 650.
Saturation curve on combined steam engine diagram, 665.
Saving due to cylinder jackets and intercooler in multistage compression, 970.
Scale of indicator springs, determination of, 608.
Schaeffer & Budenberg recording gauge, 186.
Schaeffer & Budenberg tachometers, 229.
Scleroscope for testing hardness, 137.
Secondary stresses, definition of, 63.
Seger cones, 206.
Separating calorimeter, formula for, 572.
Separating steam calorimeter, 567–572.
Separating steam calorimeters, method of use, 574.
Series producer, 854.
Shear, direct, 61.
Shear, elasticity, modulus of, 61.
Shear, horizontal and vertical, transverse loading, 59.
Shear tests, 132.
Shearing stress, definition of, 51.
Shop tests, locomotives, 786.
Siebert-Kühn apparatus for gas analysis, 875.
Siemen's water meter, 386.
Single-stage impulse turbine, 810.
Slide rule, 16.
Slide rule, directions for using, 17.
Slip in air compression, 963.
Slip in pumping engines, 775.
Smoke determinations, 540.
Soot, determination of in fuel gas, 885.
Specific gravity of cement, 158.
Specific heat, definition of, 326.
Specific heat, instantaneous, definition of, 335.
Specific heat of gases, 862–868.
Specific heat of gases, table of, 328
Specific heat of oxygen, nitrogen, hydrogen, etc. (see specific heat of gases).
Specific heat of vapors, 335.

Specific heat, mean, definition of, 335.
Specific weight of gases, 855.
Specifications, standard, for materials (see under name of society or association issuing same).
Speed indicators, 229.
Speed variation within a revolution, 235.
Standard coal, locomotive test, 787.
Standard method of conducting duty trial of pumping engines, A.S.M.E. Code, 777.
  Arrangement and use of instruments and other provisions for test, 782.
  Data and results, 784–786.
  Directions for obtaining feed-water temperatures, 778.
  Directions for measurement of feed water, 778.
  Leakage test of pump, 783.
  Main duty trial, 780.
  Test of feed-water temperatures, 777.
Standard method of conducting efficiency tests of locomotives, A.S.M.E. Code, 788.
  Preparations for test, 788.
  Road test, 795.
  Shop test, 791.
Steam accounted for by indicator diagram, 653.
Steam calorimeters, classification of, 545.
Steam calorimeter, condensing, 548.
Steam calorimeter, condensing, water equivalent for, 549.
Steam calorimeters, comparative values of, 578.
Steam chest and steam pipe diagrams, 675.
Steam consumption of perfect engine, 731.
Steam, dry saturated, definition of, 543.
Steam engine, capacity test, 726.
Steam engine, Carnot efficiency, 729.
Steam engine, cylinder efficiency, 729.
Steam engine, economy test, 726.
Steam engine, efficiency standards, 728.
Steam engine, heat balance, 731.
Steam engine indicator,
  Bachelder, 588.
  Crosby, 586.
  Crosby continuous with external spring, 591.
  Starr with external spring, 591.
  Tabor with external spring, 590.
  Thompson with external spring, 590.
Steam engine indicator, apparatus for testing various adjustments, 615–620.
Steam engine indicator, attachment to cylinder, 620.
Steam engine indicator, care of, 636.
Steam engine indicator, continuous, 593.
Steam engine indicator diagram, form of expansion line, 647.
Steam engine indicator diagram, general discussion of events, 642–645.
  Admission line, 643.
  Admission pressure, 644.
  Atmospheric line, 643.
  Back pressure line, 644.
  Boiler pressure line, 643.
  Clearance line, 643.
  Compression line, 644.
  Cut-off pressure, 644.
  Exhaust line, 644.
  Expansion line, 644.
  Initial pressure, 644.
  Mean back pressure, 644.
  Mean forward pressure, 644.
  Point of admission, 643.
  Point of cut-off, 643.
  Point of exhaust closure, 644.
  Point of release, 644.
  Ratio of expansion, 645.
  Release pressure, 644.
  Steam line, 643.
  Terminal pressure, 644.
  Vacuum line, 643.
  Wire drawing, 645.
Steam engine indicator diagram, measurements from, 645.
Steam engine indicator, early forms of, 582.
  Richards, 583.
  Watt & McNaught, 582.
Steam engine indicator, electric attachments, 635.
Steam engine indicator, errors in, 594–597.
Steam engine indicator, methods of connecting up, 632–635.
Steam engine indicator, piping for, 620.
Steam engine indicator, uses of, 580.
Steam engine indicators, small piston, 592.
Steam engine indicators, large and small, 593.
Steam engine indicators with external springs, 588.
Steam engine, radiation loss, 805.
Steam engine, regulation tests, 727.
Steam engine, testing of (see rules for conducting engine tests).
Steam engine testing, calorimetric methods, 799.
Steam engine, thermal efficiency, 728.
Steam engine, thermodynamic efficiency, 729.
Steam meter,
  Burnham, 454.
  Eckhardt, 457.
  Float, 460.
  Gehre recording, 459.
  General Electric Co., 455.

Steam meter,
  Hallwachs', 458.
  Orifice, 458.
  Pitot tube, 455.
  Sargent, 462.
  St. John, 461.
Steam meters, classification of, 453.
Steam, properties of, tables, 1092-1098.
Steam sampling for quality determinations, 546-548.
Steam, superheated, definition of, 544.
Steam tables, 1092-1098.
Steam tables, use of, 337.
Steam turbine,
  Curtis, 812.
  DeLaval, 810.
  Impulse type, 808.
  Kerr, 816.
  Multistage impulse, 812.
  Multistage reaction, 812.
  Pelton type, 816.
  Pressure stage, 812.
  Rateau, 812.
  Reaction type, 808.
  Single stage, 810.
  Sturtevant, 816.
  Velocity stage, 812.
  Westinghouse, 815.
Steam turbine losses, separation and estimation, 822.
Steam turbine, power input, 819.
Steam turbine testing (see testing of steam turbines).
Steam turbine tests, correcting results for standard conditions, 824.
Steam turbine, types of, 808.
Steam, wet saturated, definition of, 543.
Steel-yard dynamometer, 296.
Stirling cycle, 348.
Strain, measurement of, 99
Strength of materials, definitions, 51.
Stress, definition of, 51.
Stress-deformation diagrams, 53.
Stress intensity, definition of, 51.
Stress, primary, definition of, 63.
Stress, secondary, definition of, 63.
Strohmeyer extensometer, 101.
Sturtevant turbine, 816.
Submerged orifices, 361.
Subsurface or submerged floats, 398.
Suction gas producer, 851.
Sulphur determination in fuels, 509.
Sulphur dioxide, properties of, table, 1108.
Superheating of vapors, 335.
Superheating steam calorimeter, 558.
Surface floats, 398.
Suspended planimeter, 31.
Sweet measuring machine, 45.

TABLE of Beaumé hydrometer scale with corresponding specific gravities, 1091.
Tables of coefficient of discharge for submerged orifices in thin plates, 363.
Table of coefficients of friction, 1090.
Table of constants for principal gases, 857.
Table of constants for principal hydrocarbon gases, 858.
Table of formulas for beams, 60.
Table of horse power per pound of mean pressure, 1099.
Table of mean specific heat for gases, 469.
Table of melting points and specific heats of metals, 207.
Table of properties of ammonia, 1107.
Table of properties of $CaCl_2$ brine, 1110.
Table of properties of carbon dioxide, 1108.
Table of properties of NaCl brine, 1100
Table of properties of sulphur dioxide, 1108.
Table of relation between pressure in pounds per square foot, inches of mercury, etc., 366.
Table of specific gravity of oils, 247.
Table of theoretical water rates for steam engines, 1105.
Table showing ammonia absorption in water, 1109.
Table showing material tests required, 138
Tachometer,
  Autographic, 230
  Centrifugal, 229.
  Frahm, 231.
  Schaefer & Budenberg, 229.
  Veeder, 230.
Tachometers for measurement of water, 402.
Tagliabue cold test apparatus for oils, 253.
Tangential stress, definition of, 51.
Tangential water wheel, 1049.
Tank meters, 383.
Tar, determination of, in fuel gas, 885.
Temperature, definition of, 193.
Temperature-entropy diagram for gas, 685.
Temperature-entropy diagram for steam, 341.
Temperature rise during adiabatic air compression, 957.
Temperatures obtained in combustion, 468.
Tensile strength of cements, 163.
Tension loading, formulas for, 55.
Tension test machines (see testing machines).
Tension tests, character of fracture, 123.
Tension tests, directions for, 118.

Tension tests, equivalent elongation, 123.
Tension tests, form of test piece, 115.
Tension tests, laying off gauges, 119.
Tension tests, report on, 124.
Tension tests, wedge adjustments, 120.
Test pieces for compression testing, 125.
Test pieces for special materials,
  Cast iron, 117.
  Chains, 118.
  Hemp rope, 118.
  Wire rope, 118.
  Wood, 117.
Test pieces for tension testing, form of, 115.
Test pieces for torsion testing, 129.
Test pieces for transverse testing, 126.
Test specimens for cement testing, 161.
Tests and specifications, special, 138.
Tests for building stone and brick, 152.
Tests of cements and mortars, 156
Tests of paving materials, stones, ballast, natural and artificial, 155.
Testing machine accessories, 99.
Testing machines, 67.
Testing machines, cement,
  Fairbanks, 96.
  Olsen, 99.
  Riehlé, 99.
Testing machines, tension, compression and transverse, compound lever machines, 70.
  Differential lever machines, 69.
  Direct-acting hydraulic, 71.
  Emery, 72, 78-82.
  Fairbanks, 70
  Frame of, 75.
  General types of, 68.
  Hydraulic press, 67.
  Kellogg-Johnson, 71.
  Maillard, 72.
  Olsen, 70, 86.
  Power system of, 76
  Riehlé, 70, 84.
  Riehlé hydraulic, 70.
  Shackles and clamps, 68, 76.
  Simple lever machines, 69.
  Thomasset, 69
  Thurston, Polmeyer, 67.
  Weighing system of, 74.
  Werder, 71.
  Wickstead, Martins, Buckton, 69.
Testing machines, torsion,
  Olsen power torsion, 91.
  Riehlé power torsion, 90.
  Thurston, 86.
Testing of belts,
  Coefficient of friction, 323.
  Efficiency of transmission, 323.
  Initial tension, 321.
  Methods and computations, 320.
Testing of boilers (see rules for conducting boiler trials).
Testing of boilers, notes on rules, 710-724.
  Ash determination, 718.
  Boiler efficiencies, 719-721.
  Boiler leakage, 712.
  Factor of evaporation, 722-724.
  Graphical log, 715.
  Heat balance, 721.
Testing of cements,
  Constancy of volume, 163.
  Fineness, 159.
  Normal consistency, 159.
  Specific gravity, 158.
  Strength, 163.
  Time of setting, 160.
Testing of centrifugal fans, 986.
  Determination of humidity, 951.
  Measurement of velocity and pressure, 987.
  Power input, 986.
  Power output, 987.
  Scope of test and report, 988-991.
  Volume of gas delivered, 987.
Testing of centrifugal pumps, 1077.
  Head pumped against, 1078.
  Length of run, 1078.
  Power input, 1077.
  Power output, 1078.
  Quantity of water pumped, 1077.
  Scope of tests and report, 1079.
Testing of gas engines, 905
  Apparatus required, 907.
  Calibration of instruments, 906.
  Determination of principal dimensions, 906.
  Duration of tests, 906.
  Examination of engine, 905.
  Observations and calculations, 907-913.
  Report, 913-919
  Starting and stopping, 906.
Testing of gas producers, 891.
  Conducting trial, 893.
  Data and results, 895-899.
  Duration of trial, 892.
  Object of test, 891.
  Observations and calculations required, 893.
  Starting and stopping, 892.
Testing of hot air engines, 933.
  Analysis of fuel, 934.
  Analysis of waste gases, 934.
  Computations and report, 934.
  Heat balance, 935.
  Indicated horse power, 933.
  Measurement of fuel used, 933.
  Other observations, 934.
  Pump horse power, 933.

Testing of hydraulic ram, 1040.
Testing of injectors, 835.
Testing of insulation, 1011.
Testing of locomotives (see standard method of conducting efficiency tests of locomotives).
Testing of materials, methods of, 115.
Testing of oils,
  Acid, 245.
  Adulteration, 243.
  Burning point, 256.
  Chill point, 254.
  Density, 246.
  Durability, 267.
  Flash point, 254.
  Friction, 256.
  Gumming and drying, 245.
  Viscosity, 248.
  Volatility, 256.
Testing of piston air compressor,
  Air horse power, 973.
  Arrangements for, 973.
  Computations for test, 977
  Power input, 973.
  Report, 977–980.
  Weight of volume of air handled, 973–976.
Testing of pumping engines (see standard method of conducting duty trials of pumping engines).
Testing of rotary pumps, 1077.
Testing of steam boilers (see rules for conducting boiler trials).
Testing of steam engines (see rules for conducting steam engine tests).
Testing of steam pumps (see standard method of conducting duty trials of pumping engines).
Testing of steam turbines,
  Clausius efficiency, 821.
  Computations, 819.
  Cylinder efficiency, 821.
  Electrical horse power, 819.
  Internal horse power, 819.
  Potential efficiency, 821.
  Thermal efficiency, 820.
Testing of vapor compression refrigerating machines, 1028.
  Compressor and prime mover, 1029.
  Computation of results, 1033.
  Condenser, 1030.
  Cooler, 1030.
  Heat balance, 1033.
  Report, 1031,1032, 1034, 1035.
Testing of water motors, 1062.
  Length of run, 1065.
  Power developed, 1064.
  Report and computations, 1065.
  Speed and gate opening, 1065.
  Supply head, 1063.
Testing of water motors (*continued*),
  Volume or weight of water supplied, 1063.
Thalpotasimeter, 201.
Thatcher rule, 18.
Theoretical head produced by fan, 981.
Theoretical velocity of steam from the heat chart, 447.
Theoretical velocity of steam in orifices, 446.
Thermocouple pyrometers, 212.
Thermometers, 193.
Thermometer,
  Air, 199.
  Alcohol, 202.
  Electric resistance, 210.
  Employing liquids, 201.
  Employing solids, 205.
  Gas, 196.
  Mercury, 202.
Thermometer calibration, 219.
Thermometric materials, 193.
Thermometric scales, 195.
Thermometric standards, 195.
  Primary, 196.
  Secondary, 196.
Thomas electric superheating calorimeter, 566.
Throttling calorimeter, 560–563.
Throttling calorimeter computations, graphical solution, 563.
Throttling calorimeter from pipe fittings, 565.
Throttling calorimeter, theory of, 562.
Thurston oil testing machine, directions for use, 264, 266.
Thurston oil testing machine, theory of, 261.
Thurston railroad lubricant tester, 259.
Thurston standard oil testing machine, 258.
Tieftrunk apparatus for tar determination, 886.
Time for emptying tanks, 359.
Time of setting for cements, 160.
Tonnage in refrigeration, 1028.
Torsion dynamometer for solid shafts, 312.
Torsion dynamometers, 309.
Torsion loading, formulas for, 61.
Torsion meters (see torsion dynamometers).
Torsion test machines (see testing machines).
Torsion tests, directions for, with Thurston autographic machine, 131.
Torsion tests, directions, with Riehlé or Olsen machine, 129.
Torsion tests, form of test piece, 129.
Total heat in vapors, definition of, 334.

Traction dynamometers, 292.
Transmission dynamometers, general types, 293.
Transverse loading, elastic curve, 58.
Transverse loading, formulas for, 57.
Transverse loading, neutral surface in, 58.
Transverse tests, directions for, 127.
Transverse tests, form of test pieces, 126.
Transverse tests, report on, 127.
Transverse test machines (see testing machines).
Trapezoidal weir, 377.
Triangular weir, 376.
Tumlirz equation, 344.
Turbine compressors, 945.
Turbine gas meter, 436.
Turbine pumps, 1068.
Turbines, water (see water turbines).
Two-cycle gas engines, 844.

ULTIMATE strength, definition of, 52.
Undershot wheel, 1048.
Unit of electrical power, definition of, 276.
Unit of heat, 325.
Unit of power, 275.
Unit of refrigeration, 1028.
Unit of work, 275.
Universal or combination steam calorimeter, 577.

VACUUM gauges, 185.
Vacuum process of refrigeration, 1001.
Van Winkle power meter, 305.
Vapor compression refrigerating cycle, actual, 1016.
Vapor compression refrigerating cycle, ideal, 1012.
Vapor compression system of refrigeration, 1003.
Vapor cycles, 351.
Vapor, pressure and volume changes in
  Adiabatic (isentropic), 344.
  Constant pressure, 343.
  Constant volume, 344.
Vapor pressure pyrometers, 201.
Vapor tables, 335.
Vaporization, 333..
Vaporization,
  External latent heat, 334.
  Internal latent heat, 334.
  Latent heat of, 334.
  Total heat of, 334.
Veeder tachometer, 230.
Velocity of approach in weirs, 374.
Velocity of water, measurement in channels and streams, 397.
Velocity of water, measurement in pipes and conduits, 406.
Velocity changes in turbines, 812.
Velocity water meters, 387.
Venturi meter, 369, 392.
Venturi meter, theory of, 370.
Venturi meter used for measuring gas, 443.
Venturi tube, 369.
Vernier, the use of, 20.
Vernier caliper, 42.
Vertical water wheels, 1045.
Viscosimeter,
  Gibbs, 249.
  Perkins, 250.
  Stillman, 250.
  Tagliabue, 249.
Viscosimeter, other forms of, 252.
Viscosimeter with constant head, 250
Viscosity of oils, 248
Volatility tests for oils, 256.
Volume relations in combustion, 465.
Volume water meters, 389.

WANNER pyrometer, 216.
Waste detection meters, 392.
Water equivalent for bomb calorimeter, 475, 478.
Water equivalent, condensing calorimeter,
Water meter, calibration of, 396.
Water meter capacities, 388–392.
Water meters, classification of, 385.
Water meters,
  Current, 387.
  Gem inferential, 388.
  Hersey rotary, 390.
  Hersey torrent, 387.
  Hot water, 394.
  Inferential, 387.
  Nash disc, 391.
  Open type, 386.
  Piston, 393.
  Positive, 393.
  Siemens, 386.
  Velocity, 387.
  Venturi, 392.
  Volume or capacity, 389.
  Waste detection, 392.
  Worthington, 394.
Water pressure engines, efficiency, 1044.
Water turbine, 1052.
Water turbine,
  Center vent, 1059.
  Fourneyron, 1056.
  Francis, 1059.
  Impulse, 1053.
  Jonval, 1059.
  Mixed flow, 1059.
  Parallel flow, 1059.
  Radial inward flow, 1059.
  Radial outward flow, 1056.
  Reaction, 1053.
  Risdon-Alcott, 1060.

Water turbines, classification, 1053.
Water turbines, efficiency of, 1057, 1062.
Water turbines, power of, 1057, 1062.
Water turbines, testing of (see testing of water motors).
Water turbines, theory of action of stream upon vane, 1054–1055.
Water wheel, impulse, 1049.
Water wheel, tangential, 1049.
Water wheels,
  Breast, 1047.
  Overshot, 1045.
  Undershot, 1048.
  Vertical, 1045.
Webber dynamometer, 300.
Wedge scale, 100.
Wedgewood pyrometer, 205.
Weight of vapor discharged from orifices, 449.
Weight relations in combustion, 465.
Weir measurements, accuracy of, 379.
Weir measurements, flow of water, 372
Weirs, contraction in, 373
Weirs,
  Rectangular, 372.
Weirs (*continued*),
  Trapezoidal, 377.
  Triangular, 376.
Welding test, 134.
Westinghouse brake for steam turbine tests, 287.
Westinghouse-Parsons turbine, 815.
Wet and dry bulb thermometer, 952.
Wet and dry gas meters, relative accuracy, 437.
Wet gas meters, 432.
Wet saturated vapor, definition of, 334
Wilcox water weigher, 383.
Willan's Law, 726.
Willan's line, 726.
Willis planimeter, 34.
Woltman mill, 403.
Work done by centrifugal fan, 983.
Work of compressing air, 954.
Working loads, definition of, 53.
Worthington piston meter, 394.

YIELD point, definition of, 53.

ZERO circle, planimeter, 24.

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